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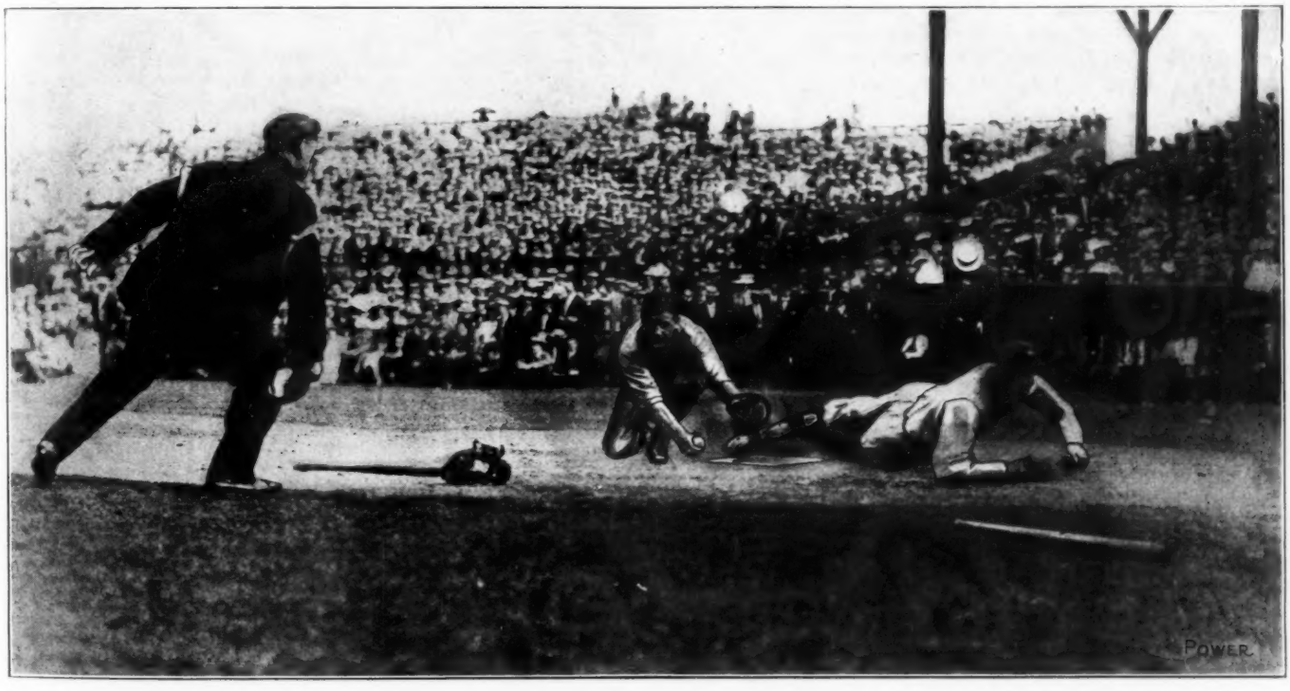
Playing the Game

AN oiler, fireman or assistant engineer is as necessary in a steam plant as a chief engineer. Neither may be as important, but each has a position to fill, either in a half-hearted way or energetically.

An oiler, for instance, may assume that his work does not count and only aims to hold his job; therefore why do any more than he has to?

One need not do more than he is compelled to, but that means a whole lot. There are two ways to work; by one it becomes drudgery and a thing detested; by the other, play, a thing enjoyed. The trouble is, most men do not know how to make their work enjoyable.



Suppose one man on a ball team lacks in interest, habitually lets chances slip without taking them. He would soon have to make way for someone who would *play the game*, a man who would work. It is each individual's work that makes the team work count.

The oiler may complain that he does not have a chance to adjust engines and handle them. True, but the successful ball player was not given a position on the ball nine untill he had shown that he was skilful. That came by practice. If the oiler wants to handle or adjust an engine, he will have to practice before he will be considered competent by the chief.

How? By lending a hand to the engineer who is doing the work, learning a little at a time. He will not have to do it, but it is *playing the game.*

No oiler plans to remain one all his life, and he will not if he *plays the game.* Steam engineering is an interesting study, and has no limit. Oiling a condenser pump is simple, but understanding its design, operation and what it accomplishes is a different proposition

A few instructions will enable one to oil an engine, but the action of the valves and of the steam in the cylinder are as greek to some men who have been accustomed to engines for years.

It is along these lines that the oiler, fireman and running engineer can make his work play. If he determines to know more about steam engineering a field of research will have been entered as interesting as it is remunerative.

He does not have to do it, and perhaps will not know that the effort is appreciated, but it is the only way to *play the game.*

Ljungström Radial Flow Turbine

Instead of the usual alternating stationary and rotating blades, moving blades only are employed. These are in circular rows upon the faces of oppositely revolving disks. The relative velocity is thus doubled and the number of stages quartered. Two generators are used which synchronize and run at the same speed without mechanical connection.

Several turbines have been proposed in which the guide blades, instead of being stationary, as in the Parsons and Rateau types, are attached to members moving in the opposite direction from the opposing blades. The complications involved by gearing these two motions to a common generator have discouraged the development of such a turbine; but the possibility of running an alternator from each of the oppositely rotating shafts and letting them synchronize, thus lock themselves together electrically, has been utilized by the brothers Birger and Frederic Ljungström, of Stockholm, who have built turbines of 500 and 1000 kw. capacity, and designed still larger units which are described in detail in *Engineering* of Apr. 12 and 19.

Allowing both sets of blades to rotate, and in opposite directions, doubles their relative velocity, which is particularly favorable to the reaction type of turbine. The Ljungströms, therefore, have made their turbine of this type. The simplest means of directing the steam upon oppositely moving blades is to use interlocking oppositely rotating disks, one connected to each motor, and this is the form which their turbine has taken. The steam enters at the center and flows radially, as shown in Fig. 1, which is a section, reproduced from *Engineering*, of the lower half of the turbine part of the 1000-kw. unit.

Steam enters through the conduits *AA*, and finds its way through the chambers *BB* and *CC* to the turbine, as indicated by the arrows. The top line of the engraving is the shaft center, and the steam, in flowing from *C* to the exhaust passage is flowing from the center toward the circumference. The section of the passage through the turbine is convergent-divergent and interrupted by rings of blading attached alternately to the disks *DD;* the first row is attached to the right-hand disk at *a*, the second to the left-hand disk at *b*, etc. The lines in the exhaust passage indicate the vanes of a diffusor by which it is claimed a lower pressure is maintained at the outlet of the last row of blading.

Fig. 2 is a section of two of the rings of blading. The blade proper is in the center at *A* and is attached to the disk *D* by the conical double bulb-headed ring, upon which the strengthening and holding rings *E* and *F* are spun. The ring *F* is held in the disk *D* by the soft iron calking strip *C*. Strips of very thin nickel *GG* run against the next ring, as shown, preventing lateral leakage.

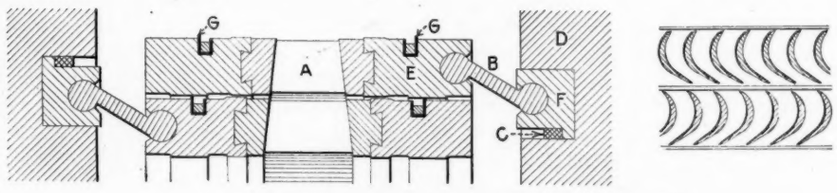


FIG. 2. SECTION THROUGH TWO ROWS OF BLADING

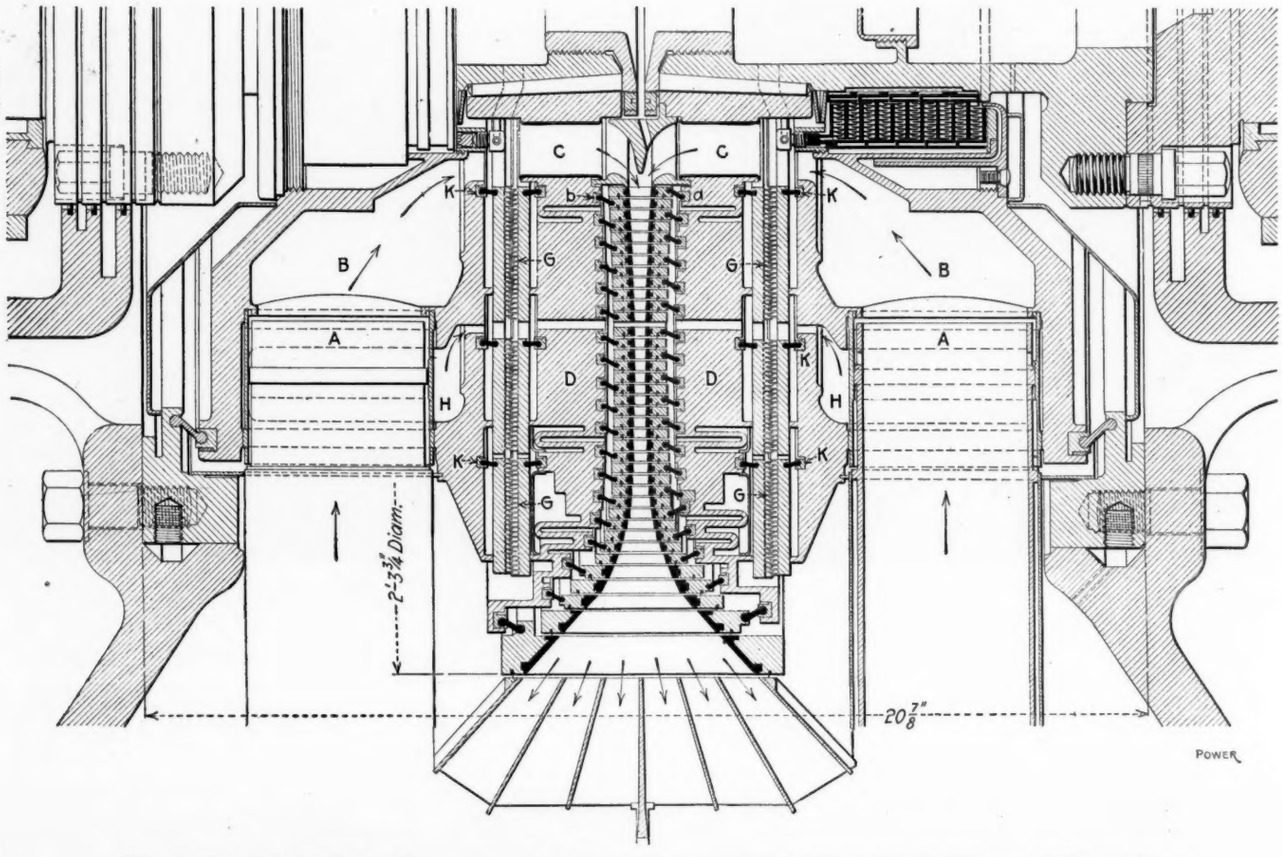


FIG. 1. THE LJUNGSTRÖM RADIAL-FLOW TURBINE. LONGITUDINAL SECTION, LOWER HALF OF 1000-KW. UNIT

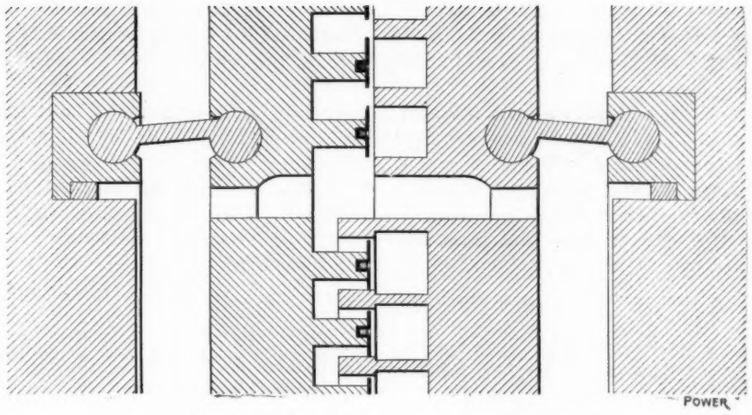


FIG. 3. DETAILS OF LABYRINTH PACKING

PARTICULARS OF TESTS OF 1000-KW. LJUNGSTRÖM STEAM TURBINE

Number of Test	1	2	3	4	5	6	7
Duration of Test in minutes	40	30	40	60	41.4	36.83	41.5
Revolutions per minute, both shafts	3000	3000	3010	3007	3018	3019	3019
Total load on generators, kw	972	788	480	1000.5	751	499.9	250.3
Total load on turbine, b.hp	1405	1150	723	1439.7	1093.2	746.2	403.7
Steam pressure before governor valve gage lb. per sq.in.	162	164	167	167	162	164	167
Steam pressure after governor valve gage, lb. per sq.in.	130	104	67	154	118	81	41
Steam temperature before governor valve, deg. F.	667	668	675	669	665	671	676
Steam temperature after governor valve, deg. F.	664	662	663	669	663	664	662
Superheat before governor valve, deg. F.	297	297	302	296	296	300	303
Superheat after governor valve, deg. F.	310	323	351	302	315	348	375
Absolute pressure in exhaust pipe, in Hg.	1.34	1.25	1.1	1.37	1.31	1.31	1.18
Absolute pressure condenser, in Hg.	1.25	1.16	1.1	1.34	1.26	1.28	1.13
Temperature of condensed steam in measuring tanks, deg. F.	56	55.4	54	42.8	41.9	41.9	41.9
Temperature of cooling water, inlet, deg. F.				39.2	39.2	39.2	39.2
Temperature of cooling water, outlet, deg.F.				57.4	53.8	50.4	45.2
Temperature of oil leaving bearings, deg. F.				126	124	127	119
Steam condensed during trial, lb	7429	4638	4123	11,550	6260	4010	2646
Steam escaping from shaft glands, lb	45	28.6	26.4	106	61.6	42	26.8
Total steam used during trial lb	7474	4666	4149	11,656	6321	4052	2763
Total steam used per kilowatt-hour	11.55	11.85	12.98	11.68	12.21	13.23	15.57
Total steam used per brake-horsepower-hour	8	8.13	8.61	8.1	8.39	8.85	9.6
Efficiency ratio, taking steam as in front of throttle valve	71.7	72.7	67.7	73.6	71.1	67.2	61.1
Efficiency ratio, taking steam as behind throttle valve	76.9	78.6	76.4	74.6	74.3	74.1	73.1

Bypass valves, either or both of which may be used, admit steam through the passages *H*, Fig. 1, for overloads. The pressure of the steam in the passage between the disks tends to force them apart and the end thrust thus produced is balanced by the dummy pistons *GG* attached to the disks *DD* by conical double bulb-headed rings similar to those

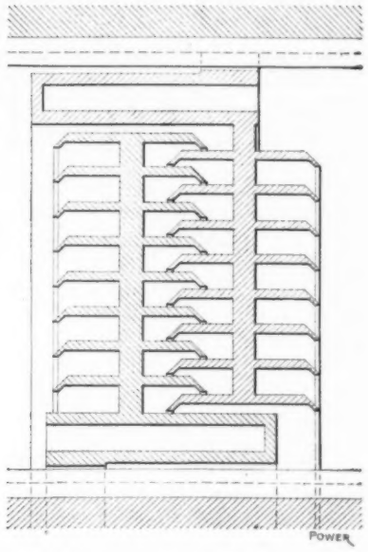


FIG. 4. LABYRINTH FOR PACKING SHAFT

used for fastening to the turbine casing at *KK*. Between them is a labyrinth packing, the details of which are shown in Fig. 3. The thin strips are of nickel, as in the case of the blades. Where the shaft passes through the casing, it has to be packed against the full initial pressure and a very effective labyrinth

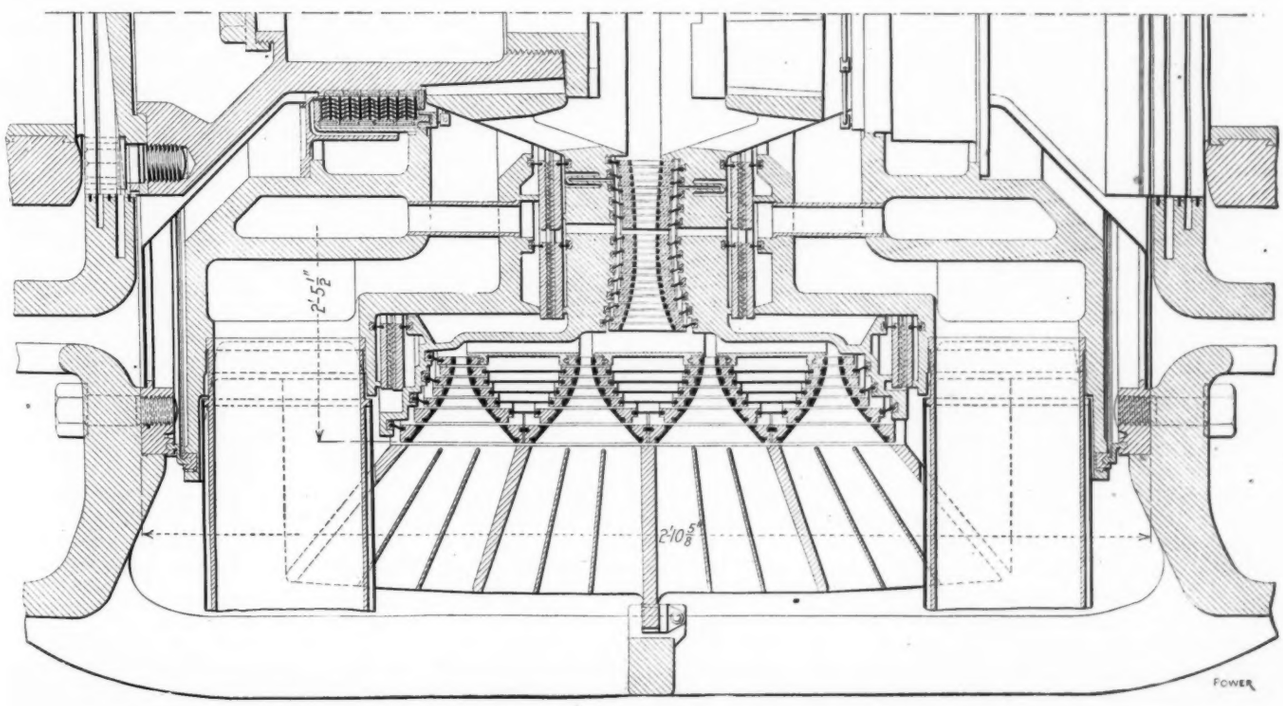


FIG. 5. SHOWING DIVISION OF LOW-PRESSURE STAGE

packing, made as in Fig. 4, is used. The rings are turned out of solid steel and designed to expand freely in all directions without distortion. The various rings are then spun over and made to just touch the rings which they embrace. A short run produces a working clearance. In 3¼ in. of shaft length they are able to get in 158 constrictions. Tests showed the leakage from both glands in the 1000-kw. machine to be but 110 lb. per hr. This leakage is led into a chamber in the side disks around the packing and thence to a feed-water heater or other point of application.

The 1000-kw. (at 3000 r.p.m.) machine is less than 17.5 ft. long, notwithstanding its two generators and exciter. Its greatest diameter is 4 ft. 2⅜ in. and its greatest height, to the top of the governor, 5 ft. 6 in. For large units the inventors propose to divide the low-pressure blading, as in Fig. 5, to get greater cross-sectional area in the low-pressure blading.

Tests of the 1000-kw. unit gave the results shown in the accompanying table. The first three were made on Oct. 10, 1911, by G. Oscar Olsson, M. I. N. A., and Axel F. Endström, of the Electriska Prüfungsanstatten, and the last four by Professors O. E. Lundström, of the Swedish Steam Boiler Association, and the Mr. Olsson above named.

Practical Features of Design

By J. E. Terman

In addition to embodying correct theoretical features a vessel should be of such design as to facilitate proper construction.

A few hints are given regarding tanks, flues and other work coming within the scope of the boiler shop.

When requiring special vessels to be built of boiler-type construction, it is best to go to a reputable boiler maker, who employs engineering talent, and let him know what is to be accomplished and allow him to get out details for the vessel. If this is not practicable or desirable, the designer should get up the design to the best of his ability and submit it to the manufacturer for criticism. In doing this, the manufacturer should be assured that his frank opinion is desired; for the experienced manufacturer invariably withholds adverse criticism of a design presented, to avoid offending a prospective customer.

Riveted joints are a part of the design which should not be attempted without practical knowledge. It may be simple to take some treatise on boiler calculations and ascertain how to proportion a joint for maximum strength, but all joints of maximum theoretical strength to resist rupture are not always practical. There should be certain relations between plate thicknesses, rivet diameters and the pitch of rivets, to produce joints that may be tightly calked and remain so in service. A design containing these qualities is imperative, regardless of the exact theoretical strength of the joint.

Openings in special vessels often cause entirely unnecessary expense in manufacture. When such openings need be only as large as an ordinary boiler manhole an 11x15-in. elliptical opening should be used; this is a standard, and reinforcements and cover plates for this shape are in nearly every boiler maker's stock. If the openings must be larger and retaining the simplicity of the manhole design is desired, the elliptical shape should be avoided, as patterns of that shape for both the opening reinforcement and the cover plate are expensive.

The shape shown in Fig. 1 has practically all of the advantages of an elliptical opening, and the circular ends simplify the pattern making. To make a pattern for the reinforcement of an opening, such as Fig. 1, a circular piece can be turned for the ends of the desired radius, and sawed in two, as indicated by the dotted lines, with straight pieces placed between these ends to get the desired length of opening. With a truly elliptical opening, the whole pattern would have to be worked out by hand at a great loss of time; the same applies to making the pattern for the cover.

When locating a connection to entirely drain a rectangular tank the connection should never be placed on the side if avoidable; a plain flat flange should be put on the bottom. Making a flange connection tight is very difficult when placed on the side in a position to completely drain the tank, as illustrated in Fig. 2. In the first place, fitting such a flange to the curve of the corner of the tank is difficult, and the rivets cannot tend to draw the flange tight against the sheet on both surfaces, which is necessary for good calking. The flange on the bottom is simpler, and it will generally be found better to sacrifice other features that might require the side connection rather than make a poor job.

A point always to be borne in mind when designing riveted vessels is that riveted joints must be calked to be made tight, and the design must be such as to make all joints accessible for calking, after the vessel has been constructed. This requirement is continually violated by the inexperienced designer in innumerable ways. Attaching a lug on top of an end seam, as illustrated in Fig. 3, is extremely bad practice, and if it is of cast iron, and cannot be calked down against the shell, making it tight is almost impossible; the head will interfere with any attempt to calk the rivet heads inside. A calking strip of plate material may be interposed between attachments of cast iron and the shell of the vessel, but this is not best, for it doubles the length of edge presented for leakage, and calking such strips tightly against both surfaces is also difficult.

If the lug is moved up, as in Fig. 4, and the supports are arranged to accommodate this position, the rivets need only be calked on the inside. Even here, having the lug of such material that it may be calked tight against the shell of the vessel, is better. Inside calking has to be done largely by guesswork, while if the external edges are calked, the effect can be noted under pressure and the leaks definitely located.

Fig. 5 shows a practically impossible construction as the girth seam under the lug cannot be calked. This seam could be made tight before attaching the lug, but there would be no certainty that attaching it would not disturb the calking edge. If the lug was broad this seam could not be reached for further calking after the lug had been attached. An almost impossible construction is represented in Fig. 6, because the longitudinal seam from *A* to *B* cannot be calked properly, due to the interference of the lug.

A designer should never specify exactly where joints should come in tank work; if they must not come at some places, it is best to designate these places and allow the manufacturer all the latitude possible. Very frequently vessels of this class are manufactured from stock material carried by the boiler shop, which is usually in standard-size sheets, and to make a tank with the seam locations specified generally requires a waste of plate, for which the purchaser has to pay.

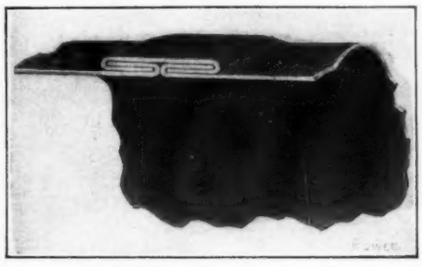
Often a vessel is designed for use as a receiver, such as shown in Fig. 7, with only small outlets for pipe connections. A good boiler maker can usually do many things that are seemingly impossible, but with a vessel of considerable size and small openings, riveting in one of the heads in a design, such as Fig. 7, is practically impossible. If there are reasons why an opening of suitable size to work through cannot be placed in such a vessel, one of the heads should be backed in as illustrated in Fig. 8.

The average designer of smoke flues who has not had shop experience usually attempts to produce a design which will theoretically offer the least resistance to the passage of the gases and be of proportionate area at every point to the volume of gas to be handled. This, from the standpoint of use alone, is commendable, but some consideration should be given to first cost and practical construction. A flue such as Fig. 9, although actually containing less metal than one like Fig. 10, is usually more costly, and the difference in the efficiency between the two would be very small. In rectangular flues, the thickness of stock and the spacing of the angle-stiffeners is important in insuring satisfactory performance in service, and only one who has had practical experience in such matters should be consulted to determine what is best.

Where two branches of a rectangular flue are brought together, as in Fig. 11, two pieces of sheet-iron baffle, bent and fastened in the flue at *CC*, afford an easy passage for the gases, prevent eddy currents and make a much stronger and cheaper flue to construct than that shown in Fig. 12. Straight lines should be followed as far as practicable in flue design, as every twist and change of shape presents more or less difficulty to the boiler maker and tends to prevent making a tight, strong structure which is essential to good service and can usually be constructed at less expense than some special shape.

Interlocking Spiral Pipe

The Standard Spiral Pipe Works, 25 North Dearborn St., Chicago, Ill., is placing on the market a reinforced galvanized spiral pipe, made of two steel strips of different widths and with a continuous interlocking seam. Both strips are rolled to shape on the edges and are interlocked and rolled down under very high pressure.



REINFORCED SPIRAL PIPE

The strips are first cut to the required width from long sheets and the ends are welded together as illustrated. The strip used for the reinforcing band comes in long lengths. These strips are fed into the machine at one end, rolled down and interlocked in the machine, coming out at the other end in the shape of pipe.

This pipe is smooth on the inside, with the reinforcing band running on the outside the entire length of the piece, and can be made endless. Ample folds are made at both strips so that a large bearing surface or contact is obtained at the seam. Where the reinforcing band passes around the pipe there are four thicknesses of metal. The pipe is supplied galvanized in any length up to 20 ft., and asphalted in any length up to 25 ft.

A slip-sleeve joint is made with a seamless sleeve welded into one end of the pipe. Lugs are also welded on the outside of both ends of the pipe. In making connections one end is slipped into the other and the pipes are then drawn together by wires over the lugs and a joint is obtained capable of withstanding a water pressure up to 60 lb.

Another style of connection is made with forged-steel flanges electrically welded on the ends of the pipe and furnished with bolts and gaskets complete. This pipe can also be supplied with cast-iron or forged-steel bolted joints. The pipe requires no soldering except where the ends are drawn together.

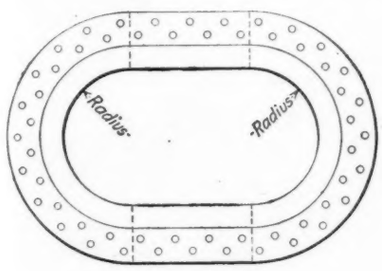


Fig. 1

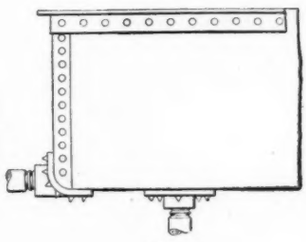


Fig. 2

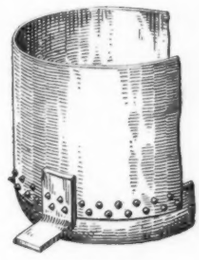


Fig. 3

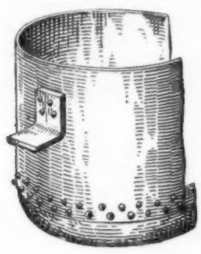


Fig. 4

Fig. 5

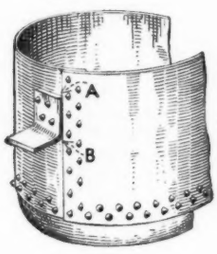


Fig. 6

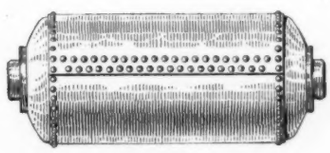


Fig. 7

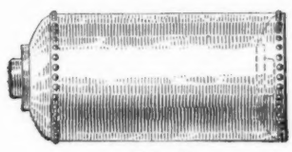


Fig. 8

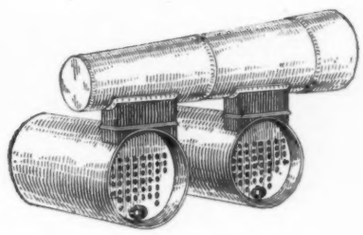
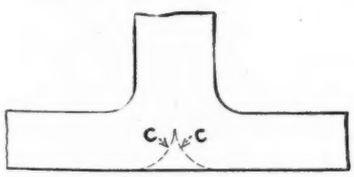
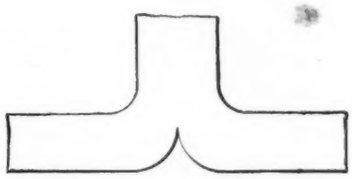


Fig. 9



Plan, Fig. 11


Plan, Fig. 12

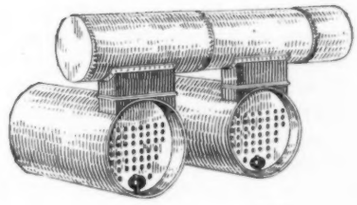


Fig. 10

Transporting a Boiler in the Tropics

By C. Carleton Semple

Describes the difficulties encountered in transporting an 80-hp. return-tubular boiler to a mine. Wooden walls, cement and fire brick were used for the boiler setting. Later a sheet iron casing replaced the wooden wall.

Several years ago the writer went to Nicaragua to take charge of a gold mine 90 miles from the coast and about one mile back from a small river, navigable only by shallow-draft dugout canoes or pit-pans. The river ran into the Caribbean at Prinzapolka, about 90 miles north of Bluefields, the nearest port of call of a line of fruit steamers.

The mine was in the midst of a dense forest, at low elevation, and in a part of the country where rains were heavy for nine months of the year. The road from the mine to the river bank was usually so muddy that one would sink to the knees in soft clay. Two rapids in the river added to transportation difficulties, and in places the water was so shallow that the canoes had to be dragged over smooth rock ledges in the river bed.

New Boiler Required

The mine steam plant consisted of a 30-hp. locomotive boiler and a slide-valve engine. Green wood fuel was used, because, due to the peculiar climatic conditions, if left to dry out and season the wood would rot. The firebox being too small to satisfactorily burn green fuel, and the tubes badly scaled, the boiler could not supply half the steam for which it was designed.

As it was intended to rebuild the mill and triple the capacity a new 80-hp. return-tubular boiler was ordered, which, with the old boiler retubed, would generate all the steam needed. A return-tubular boiler was desirable as it permitted enlarging the firebox to any desired extent, no skilled bricklayers are required for its setting and it is easily cleaned. The last feature was important as the water was bad.

An 80-hp., 60-in. by 16-ft. return-tubular boiler weighs over 11,000 lb., which was almost double the weight of the heaviest single piece of machinery that had theretofore been taken into that part of the country. The largest canoe on the river was about 45 ft. long, with an 8-ft. beam and capable of carrying 14,000 lb. Such a load had often been carried, but never in a single piece, and never with the center of gravity high above the gunwales. The transportation was considered feasible and, although the contractors would not guarantee safe delivery, the job was undertaken at the usual tariff of $2.20 per 100 lb. from Bluefields to the mines.

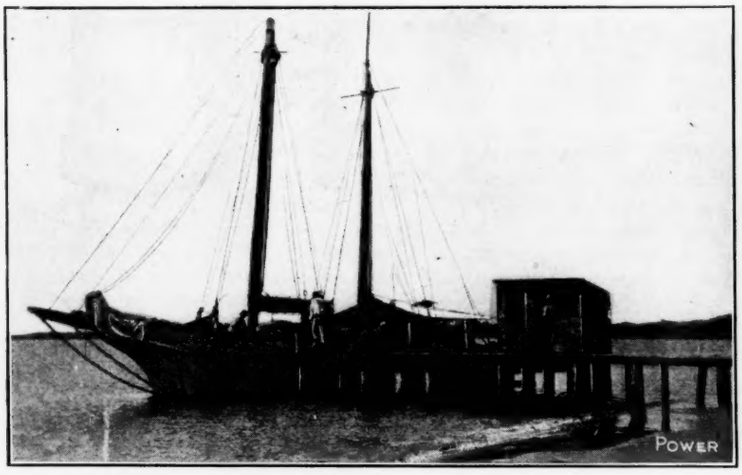


Fig. 1. Arrival of the Boiler at Prinzapolka

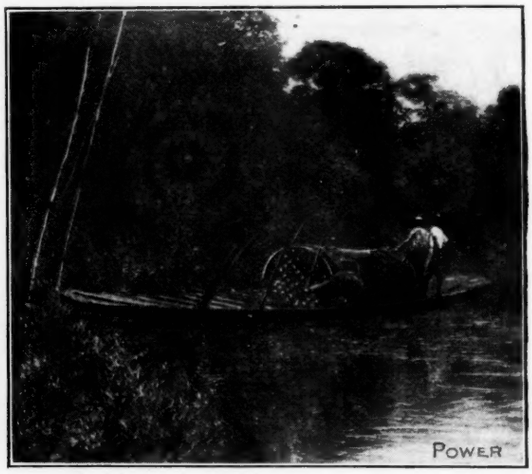


Fig. 2. At the River Bank, 25 Ft. High

Transporting the Boiler

When the boiler arrived at Bluefields it was loaded on one of the small gasoline auxiliary schooners of from 35 to 50 tons burden which ply the coast between Bluefields and Prinzapolka, and was placed athwartship just abaft the foremast, Fig. 1. After a calm night on the sea it arrived at Prinzapolka the following morning, where it took a day's hard work to remove the boiler from the schooner to the dock and from the dock into a canoe manned by 17 Mosquito indians, and 17 days later reached the mine. At Walpatara Rapids, several ropes were let out from the bow and on each side amidships, and the canoe was warped over the shallows, as shown in Fig. 3.

The river bank was fully 25 ft. high at the mine, Fig. 2. In hoisting the boiler up this bank, four heavy, wooden ship blocks, 100 ft. of 2-in. and 500 ft. of 1-in. manila rope were used. A jib crane was rigged up at the river bank, using a stout ironwood boom and a strong growing tree as a mast, Fig. 4. It was intended to sling the boiler with 2-in. rope and raise it by a ⅝-in. steel rope, using half a dozen yoke of oxen to hoist the boiler from the canoe to the level of the bank and then swing it in. This plan was changed when the steel rope "turned up missing" with only 1-in. rope to take its place. Four blocks and falls were rigged with this small rope and four teams of oxen of two yokes each were used, one team on each rope. They managed to lift the boiler out of the canoe, but could not be made to pull in unison and the scheme had to be abandoned. The 125 mine men were called to man the ropes and the boiler was gradually raised to the top of the bank after an entire morning's work.

More Difficulties

To haul a boiler of such weight over a mile of bush road, so muddy that the oxen sank to their knees, was a task. A wheeled vehicle was out of the question; skids alone could be used. The forest was searched for a tree that forked near the ground into two branches of equal diameter. When found, the small branches were trimmed off and the trunk cut 2 ft. below the fork, and the end of each branch at a distance of 18 ft. from the fork.

This gave a V-shaped sled, which, after trimming the front or trunk end upward, and smoothing the bottom side of the branches, readily slid along the muddy

road. The ends of the two branches were drawn together with a tie-rod, and cross-pieces spiked on the upper sides. The boiler was loaded on these cross-pieces and held in place by triangular blocks nailed to them. A snatch-block rigging and two yoke of oxen brought the boiler to the mine in about six hours.

Wood and Cement Boiler Setting

At this particular mine the cost of freight on bricks would have been 3c. per lb. The writer had once before obtained satisfactory results in setting a boiler in concrete, lining the walls with firebrick, and resolved to use the same setting here. The firebricks were laid up in a claywash, forming the inner wall of the setting. The outer form was built of 2-in. plank and concrete filled in between the two walls. If cement is shipped in galvanized-iron drums it can be used as in the temperate zones, but unfortunately the cement used for the boiler walls was shipped, contrary to instructions, in sacks and barrels. Cement traveling for several weeks through a damp, hot climate greatly deteriorates and often becomes practically useless.

Naturally the boiler walls did not last long, and in a few months began to crumble and fall away. Finally they were removed altogether and it became a question of what material could be used in place of brick or masonry. Hard stone suitable for stone walls was not to be found in the vicinity. It was, therefore, decided to build a wooden frame around the sides and back of the boiler and as high as the old concrete wall had been. Then, between the firebrick lining and this wooden wall powdered clay, that had been dried in the sun, was tamped, which made a fairly good backing for the firebrick lining, and as the space between the wooden walls and the lining was 8 or 10 in., the heat reaching the wooden walls was not sufficient to cause charring. This setting is shown in Fig. 5.

Had the clay been placed between the brick lining and the wooden walls while in a moist condition, trouble would soon have been experienced from cracking as the moist clay dried, which would have permitted the board walls to burn and would also have admitted so much air as to interfere seriously with the combustion. The wooden walls were used for two years before they had to be replaced with some old sheet-iron plates, which had no use. These were riveted together and used instead of the wooden walls. Some trouble was experienced with this setting because the heat that did penetrate the clay caused warping.

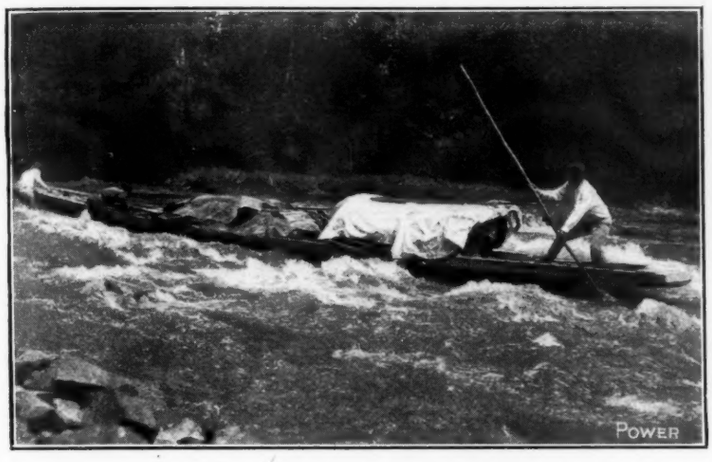


Fig. 3. Warping Over the Rapids

Fig. 4. Unloading the Boiler

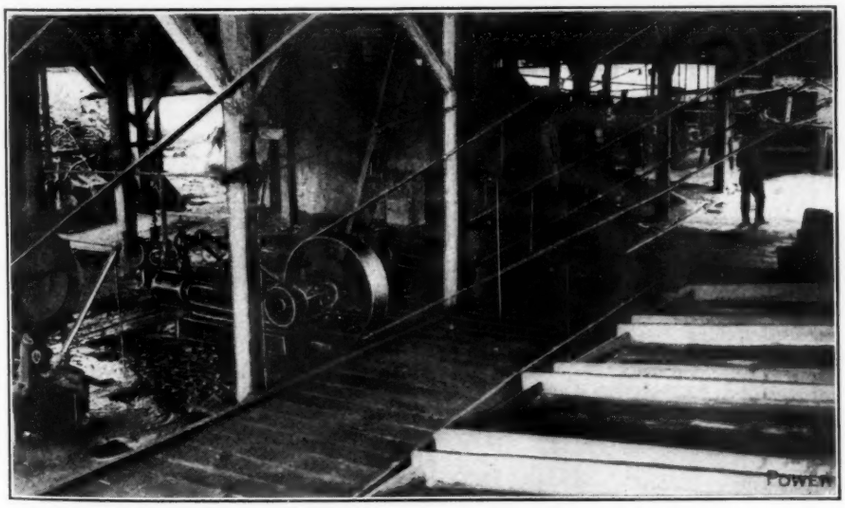


Fig. 5. View of the Steam Plant, the New Boiler and Wooden Setting Beyond the Engine

The cost of this boiler, as installed was about $2250. The boiler cost $900; the cost of transportation to the mine and setting made up the remainder. This is at the rate of approximately $28 per hp.

Electrical Disturbance on Shipboard

The growing use of electricity on board ships has had such a disturbing effect on the accurate working of chronometers that Capt. Cust, hydrographer to the British Navy, recently requested the assistance of the magnetic department at Greenwich Observatory in discovering how best to counteract the influence of a vessel's dynamos on its Greenwich time indicators. The experiments at Greenwich have been successful, and some remarkable variations in the magnetic effects have been discovered, according to the position in which the balance of a chronometer or watch is placed relative to the points of the compass. It was also found that a chronometer in the vicinity of a magnetic field is quite unaffected if its balance moves through a regular arc of 210 deg.

Mexico has granted a 99-year concession to an American company which permits taking water from the Santa Maria and Gallinas rivers for the generation of electricity.

U. S. Census Distribution of Power

By Davis H. Tuck

Ninety per cent. of the 18,760,686 hp. generated in the United States in 1909 was developed by manufacturers. Its distribution is given in the tables. Steam is the largest power source. The use of gas engines and electric power has strikingly increased.

In 1909, it is reported in the thirteenth United States census, 18,760,686 primary horsepower was generated or rented by manufacturing establishments in the United States. Table 2 shows the distribution.

Ninety per cent. of the horsepower in 1909 was that of engines or motors owned by the manufacturing establishments themselves, and 10 per cent. was rented power, mostly electric. Especially striking is the increased use of gas engines and of electric power, both that rented from outside concerns and that generated by the manufacturers. The total horsepower of electric motors in 1899, including those operated by purchased current and by current generated in the establishment, was 492,936; in 1909 it was 4,817,140, nearly ten times as great.

Renting electric power is rapidly becoming more common among small establishments, while there is a tendency for the large concerns to use electric motors for applying power which they themselves generate.

The largest amount of electric power is used by the steel works and rolling mills, the next largest being the foundries and machine shops. In the former the electric power is a little over one-third of the total amount of primary power and in the latter nearly three-fourths.

Table 1 shows, in geographic divisions,

TABLE 1

Division and State	Total horsepower (excluding duplication)	Owned by Establishments Reporting					Rented		Electric Motors	
		Steam engines	Gas engines	Water wheels	Water motors	Other	Electric motors	Other	Total [1]	Run by current generated by establishment
United States	18,680,776	14,202,137	754,083	1,807,144	15,449	29,293	1,749,031	123,639	4,817,140	3,068,109
Geographic divisions:										
New England	2,715,121	1,656,911	44,451	751,270	3,412	2,055	218,642	38,380	663,143	444,501
Middle Atlantic	5,531,502	4,151,662	274,274	466,541	3,947	11,736	568,723	54,619	1,737,236	1,168,513
East North Central	4,382,070	3,491,418	283,450	206,393	2,048	4,766	375,876	18,119	1,297,447	921,571
West North Central	1,101,990	838,988	57,434	82,791	3,539	939	115,002	3,297	266,534	151,532
South Atlantic	1,837,401	1,434,221	36,688	184,431	1,082	5,321	171,146	4,512	343,393	172,247
East South Central	1,036,560	953,511	12,270	29,040	275	1,690	38,580	1,194	108,409	69,829
West South Central	873,350	805,640	29,291	3,060	48	2,513	31,807	991	78,893	47,086
Mountain	400,766	306,786	4,188	21,345	198	224	66,956	1,069	113,984	47,028
Pacific	802,016	563,000	12,037	62,273	900	49	162,299	1,458	208,101	45,802
New England:										
Maine	459,599	168,595	6,583	253,830	1,912	179	27,203	1,297	54,266	27,063
New Hampshire	293,991	139,128	1,238	127,490	521	30	21,209	4,375	45,351	24,142
Vermont	159,445	64,252	2,160	78,881	181	415	12,917	639	21,233	8,316
Massachusetts	1,175,071	834,701	18,326	185,996	520	895	109,996	24,637	402,492	292,496
Rhode Island	226,740	175,293	3,300	31,376	41	39	13,697	2,994	42,130	28,433
Connecticut	400,275	274,942	12,844	73,697	237	497	33,620	4,438	97,671	64,051
Middle Atlantic:										
New York	1,997,662	1,080,877	99,899	394,221	1,397	3,583	389,945	27,740	689,976	300,031
New Jersey	612,293	529,668	20,867	18,558	1,118	180	33,157	8,745	182,475	149,318
Pennsylvania	2,921,547	2,541,117	153,508	53,762	1,432	7,973	145,621	18,134	864,785	719,164
East North Central:										
Ohio	1,583,155	1,362,134	103,801	15,777	330	1,586	93,592	5,935	417,844	324,252
Indiana	633,377	448,528	109,105	7,446	447	599	65,548	1,704	233,193	167,645
Illinois	1,013,071	838,199	37,025	12,178	513	1,433	117,007	6,716	398,621	281,614
Michigan	598,288	465,520	13,988	41,442	577	16	74,270	2,475	133,064	58,794
Wisconsin	554,179	377,037	19,531	129,550	181	1,132	25,459	1,289	114,725	89,266
West North Central:										
Minnesota	297,670	199,777	7,174	56,631	2,939	25	30,297	827	52,212	21,915
Iowa	155,384	121,882	8,025	6,326	85	147	18,463	456	40,736	22,273
Missouri	340,467	280,489	11,159	3,532	206	5	44,056	1,020	106,941	62,885
North Dakota	13,196	10,170	1,304	530	.....	.....	1,164	28	1,698	534
South Dakota	17,666	12,257	2,784	927	12	.....	1,683	3	2,084	401
Nebraska	64,466	44,806	4,408	7,361	75	76	7,530	210	15,942	8,412
Kansas	213,141	169,607	22,580	7,484	222	686	11,809	753	46,921	35,112
South Atlantic:										
Delaware	52,779	42,266	766	5,183	12	.....	4,502	50	17,910	13,408
Maryland	218,244	181,326	5,736	11,953	121	1,069	17,108	931	44,921	27,813
District of Columbia	16,563	12,169	1,073	775	.....	43	2,433	70	4,527	2,094
Virginia	283,928	221,303	3,664	45,122	33	38	13,356	412	42,043	28,687
West Virginia	222,896	187,389	16,952	12,901	71	.....	5,330	253	28,543	23,213
North Carolina	378,556	271,944	2,356	41,619	307	1,035	60,044	1,251	86,002	25,958
South Carolina	276,378	193,052	1,264	38,422	75	2,400	41,130	35	67,620	26,490
Georgia	298,241	240,264	3,380	28,288	460	536	23,890	1,423	44,264	20,374
Florida	89,816	84,508	1,497	168	3	200	3,353	87	7,563	4,210
East South Central:										
Kentucky	230,224	207,591	4,724	5,320	57	915	11,314	303	31,268	19,954
Tennessee	242,277	215,338	1,853	9,670	107	4	14,666	639	29,586	14,920
Alabama	357,837	328,275	4,616	13,812	111	732	10,104	187	39,928	29,824
Mississippi	206,222	202,307	1,077	238	.....	39	2,496	65	7,627	5,131
West South Central:										
Arkansas	173,088	168,152	1,374	639	35	52	2,581	255	7,417	4,836
Louisiana	346,652	331,370	3,496	65	10	2,401	9,077	233	27,139	18,062
Oklahoma	71,139	56,643	8,676	470	2	.....	5,281	67	7,887	2,606
Texas	282,471	249,475	15,745	1,886	1	60	14,868	436	36,450	21,582
Mountain:										
Montana	90,402	49,654	223	13,583	63	.....	26,504	375	27,301	797
Idaho	42,804	35,529	242	2,403	4	.....	4,606	20	8,409	3,803
Wyoming	7,628	6,467	182	456	9	.....	514	.....	801	287
Colorado	154,615	135,645	1,464	1,377	49	105	15,874	101	35,944	20,070
New Mexico	15,465	11,781	365	74	.....	.....	3,245	.....	4,586	1,341
Arizona	39,140	34,193	1,285	129	.....	19	3,314	200	15,100	11,786
Utah	42,947	28,984	226	2,926	71	100	10,592	48	15,402	4,810
Nevada	7,765	4,533	201	397	2	.....	2,307	325	6,441	4,134
Pacific:										
Washington	297,897	257,230	1,494	7,842	223	19	30,951	138	43,615	12,664
Oregon	175,019	112,244	428	47,041	397	.....	14,811	98	20,802	5,991
California	329,100	193,526	10,115	7,390	280	30	116,537	1,222	143,684	27,147

[1] Includes the horsepower of motors run by rented current and also of those run by current generated by the establishment.

the amount of each of the several kinds of power used in manufacturing industries in 1909. The relative total amount used is largely dependent upon the character of the industries predominant in each division or state, and the extent to which the different kinds are used depends upon the character of the industries and the situation of each state as to supplies of coal, petroleum and gas and the availability of water power.

In every state except Maine and Vermont, steam engines are the most important source of power. The proportion which gas-engine power represents of the total is larger in the East North Central division than in any other division, partly on account of the proximity of gas wells. The Middle Atlantic States rank next in the proportion of the total developed by gas engines.

For power obtained by waterwheels owned by the manufacturing establishments, New England ranks far ahead of the other divisions both in the absolute amount of power and in the proportion which water power represents of the total. More than two-fifths derived from waterwheels owned by manufacturing establishments is found in New England, and more than one-fourth of the power of the New England factories is derived from waterwheels. The Middle Atlantic division ranks next.

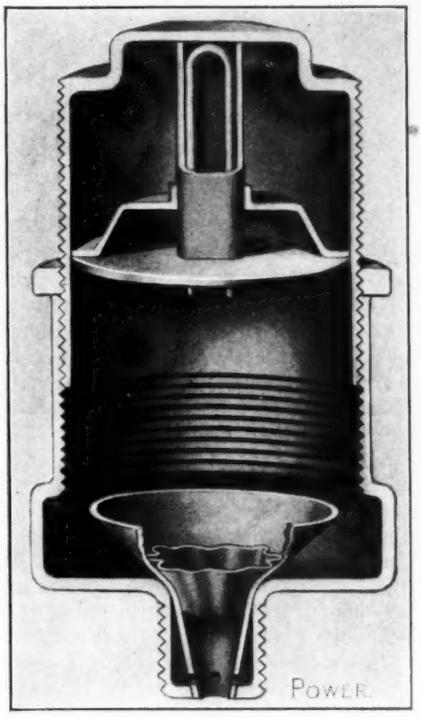
The largest absolute amounts of power of electric motors (including both those operated by current generated in the establishment and those operated by purchased current) are reported from the Middle Atlantic, the East North Central and New England divisions.

The leading individual states using gas engines to develop power are Pennsylvania, Indiana, Ohio, New York, Illinois, Kansas and New Jersey. In the use of waterwheels to develop power the states are New York, Massachusetts, Wisconsin, New Hamphsire, Vermont, Connecticut, Minnesota, Pennsylvania, Oregon, Virginia, North Carolina and Michigan.

In the absolute amount of electric power for manufacturing, Pennsylvania leads, followed by New York, Ohio, Massachusetts, Illinois, Indiana and New Jersey.

Lewis Compressed Air Grease Cup

This grease cup consists of a body with internal threads into which the top part of the cup screws. On the interior is a saucer-shaped disk, held firmly in



COMPRESSED AIR GREASE CUP

the top of the cup by an oval-shaped tube. In this tube a wire staple slides and is connected with and guides a movable disk. In the bottom of the cup is a funnel, corrugated where it comes in contact with the top of the shank. At the extreme bottom of this funnel, inside the shank, are a number of small holes. When the cup is full of grease, and the top is screwed into the body, the air is compressed in the top of the cup, which creates a pressure, or feeding force. The flat portion of the saucer-shaped cup prevents any grease passing the disk should the cup be screwed on too far. The funnel in the body directs the grease to the outlet.

After the top has been screwed down its full length and the disk rests on the funnel, the cup should be refilled, but if this is not done, the grease around the funnel is forced through the corrugations and small holes at the bottom by the air pressure passing around the disk and acting directly on the grease. This cup is manufactured by the Lewis Grease Cup Co., 1325 Arch St., Philadelphia, Penn.

The Swain Bearing Candle Lubricator

This lubricator consists of a metallic tube, open at one end, and fits into a hole in the bearing cap. The top end has a removable cap through which an indicating rod projects. This rod is fastened to a weight which rides on a lubricating candle and indicates the length of the candle in the tube. The lubricating candles are made in various degrees of hardness to suit different requirements.

The lubricator is manufactured by the Swain Lubricator Co., 328 West Lake St., Chicago, Ill.

"Twenty-nine college professors from 25 different colleges in 19 states carry their dinner pails and earn 20c. an hour in local industrial plants. They have forfeited their vacations that they might get more practical experience," says a Pittsburgh dispatch. These young "profs" have charge of the engineering departments of their respective schools. Three cheers for their spunk, and three times three for their common sense. Theory & Practice are doing business at the old stand.

TABLE 2

Power	Number of Engines or Motors			Horsepower			Per Cent. Distribution of Horsepower		
	1909	1904	1899	1909	1904	1899	1909	1904	1899
Primary power, total	408,472	231,363	168,143	18,680,776	13,487,707	10,097,893	100.0	100.0	100.0
Owned	209,163	169,774	168,143	16,808,106	12,854,805	9,778,418	90.0	95.3	96.8
Steam	153,482	127,267	130,710	14,202,137	10,825,348	8,139,579	76.0	80.3	80.6
Gas	34,352	21,515	14,334	754,083	289,423	134,742	4.0	2.1	1.3
Water wheels	20,126	19,595	23,099	1,807,144	1,641,949	1,454,112	9.7	12.2	14.4
Water motors	1,203	1,397	(1)	15,449	5,931	(2)	0.1	(2)	(1)
Other				29,293	92,154	49,985	0.2	0.7	0.5
Rented	199,309	61,589	(1)	1,872,670	632,902	319,475	10.0	4.7	3.2
Electric	199,309	61,589	(1)	1,749,031	441,589	182,562	9.4	3.3	1.8
Other				123,639	191,313	136,913	0.7	1.4	1.4
Electric motors	388,854	73,119	16,891	4,817,140	1,592,475	492,936	100.0	100.0	100.0
Run by current generated by establishment	189,545	73,119	16,891	3,068,109	1,150,886	310,374	63.7	72.3	63.0
Run by rented power	199,309	(1)	(1)	1,749,031	441,589	182,562	36.3	27.7	37.0

[1] Not reported.

[2] Less than one-tenth of 1 per cent.

Deane Motor Driven Triplex Mine Pump

Modern mining operations are carried on at such depth that the problem of unwatering the works is of interest, and special equipment has been developed to deal with its solution. The application of electrical energy in mine work simplifies the work where the cost of fuel is high and reduces the cost of operation.

The accompanying illustrations represent recent development in large mine-pumping units. The pump shown is of the horizontal, triplex, double-plunger, pot-valve type. The machine is so designed as to permit its being lowered through a shaft 3 ft. 4 in. by 4 ft., but all parts subject to wear or breakage are so segregated as to facilitate replacements.

A single reduction of double-helical or herring-bone gears, Fig. 1, transmits power from the motor to the pump crankshaft with a speed reduction from 580 to 63 r.p.m. The motor is of the two-speed type to provide flexibility in the total capacity of the station; flexible couplings are used to secure smoothness of operation. This form of drive gives an estimated mechanical efficiency of over 85 per cent. of the energy applied at the motor.

The crankshaft and connecting-rods are steel forgings. The crankshaft bearings are of the chain-oiling type. All working parts are accessible for wear, but the adjustments do not require delicate attention.

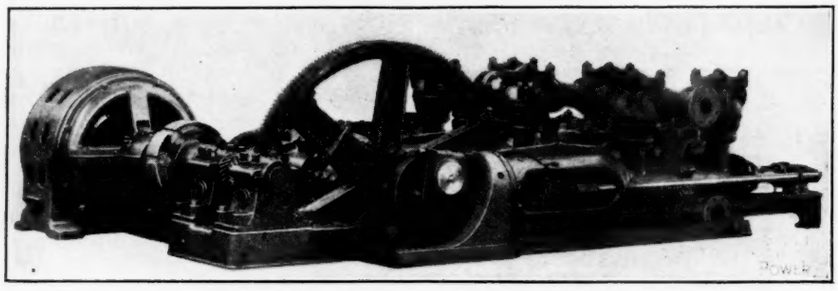


FIG. 1. MOTOR AND DRIVING GEAR

The water end of the pump, Fig. 2, is made with deep-plunger stuffing-boxes. All valves are accessible from above and the handhole covers on the pots are held by swing bolts. The suction and discharge pots are inter-changeable. The flow of the fluid is as nearly direct as is possible, thereby permitting satisfactory operation at high rotative speeds. The glands and stuffing-boxes have renewable bronze bushings.

The principal data regarding this pump are:

Diameter of plunger, 4½ in.;
Length of stroke, 12 in.;
Speed of pump, 63 r.p.m.;
Capacity of pump, 300 gal. per min.;
Working head, 715 lb., or 1650 ft.;
Motor, 175 hp., 2200 volts, three-phase, 30-cycle;
Speed of motor, 600 and 300 r.p.m.

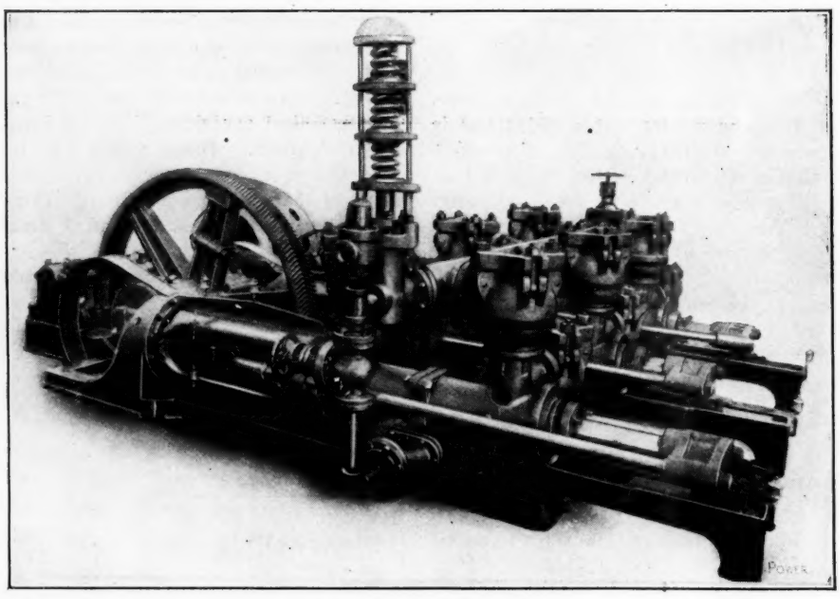


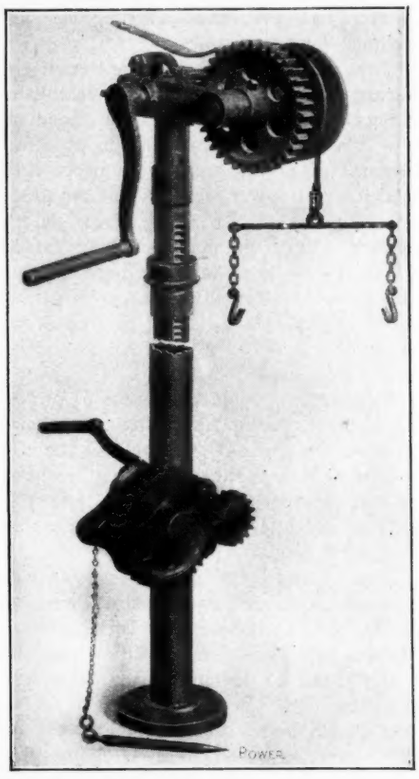
FIG. 2. WATER END OF PUMP

Two of these machines, built by the Deane Steam Pump Co., Holyoke, Mass., are being installed in a Colorado mine at the 1650-ft. level. Both machines deliver through a common column pipe to the surface.

G. & G. Ash Hoist

Loaded ash cans, barrels, boxes, etc., up to 500 lb. weight can be raised on the hoist herewith illustrated. It can be arranged so that the platform can be raised 5 or 6 ft. above the grade level, so as to empty ash cans directly into wagons without rehandling.

The hoist is adjustable, and no part of it shows above the sidewalk when it is not in use. Raising and lowering is made easy by the use of compound gearing, and a brake attachment permits the lowering of heavy loads without trouble. A safety ratchet device prevents the extension falling when being raised or lowered. Gillis & Geoghegan, 537 to 539 West Broadway, New York City, build this hoist.



G. & G. ASH HOIST

When you were a brash young fireman, did you ever scoop the other scooper's scoop when your own looked like the ragged edges of Despair and the other fellow's like a soft day's labor? A Western railway claims to get better service from its firemen with the individual scoop. How do you think such an arrangement would work out in a boiler room?

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Determining Neutral on Interpole Machines

By J. S. Kinney

To operate satisfactorily, an interpole motor or generator must have its brushes set exactly on neutral; if not, the commutation will be poor. A method for putting the brushes on neutral, which has recently come into favor, especially on large machines, is to separately excite the interpole field, and shift the brush-supporting ring until there is no voltage between adjacent brush arms.

The procedure is as follows: The spacing of the brushes is first checked by wrapping a narrow strip of paper around the commutator under the brushes and marking along the front of each brush. If there is more than one brush per arm, the spacing is checked for one brush on each arm, and the other brushes on the arm are checked to this one. After removing the paper, the distance between the marks are measured, and if not equal, they should be made so by adjusting the brush arms or holders. All the brushes are raised and in one holder on each arm is placed a brush which has had the contact edge beveled, care being taken to grind off the same amount on each side.

The machine is then driven at approximately its rated speed, by a motor, or other means.

With no field on the machine, a low reading voltmeter, say 15 volts full scale deflection, is connected across two adjacent brush arms to measure the voltage due to residual magnetism. This residual magnetism must be reduced to zero by separately exciting the shunt field through a high resistance and closing the field switch for an instant. The voltmeter should be removed from the circuit while this is being done, as the inductive kick would be likely to injure the meter. If, on replacing the voltmeter in circuit, the voltage due to residual magnetism is found to be greater than before, it is evident that the current through the field was in the wrong direction, and will have to be reversed.

After the residual field has been reduced to zero, the interpole field is separately excited with from 2 to 4 per cent. full-load current, and the reading of the voltmeter connected across the arms is noted. The interpole field is then removed and the residual magnetism, if there is any, is reduced again. The interpole field is again excited, with the same strength as before, and the brushes shifted until the voltmeter reading is practically zero. The residual magnetism must be kept down to nearly zero while the brushes are being adjusted. The final adjustment should be made with a voltmeter having a full-scale deflection of about 1½ volts.

The theory underlying this method is that, when the interpoles are excited, and the brushes on neutral, equal and opposite voltages are induced in the armature conductors under each pair of poles.

This method has the advantage over the "kick" method, in that an average neutral position for all the brush arms is found, while by the "kick" method, the neutral must be found for each arm separately, and a guess made as to the correct average position.

Edison Bipolar Machines as Railway Boosters

By L. J. Freed

Edison bipolar generators are being replaced by modern multipolar machines, and are fast going to the scrap pile. This has been their fate in the plant where the writer is employed, except four, which have been utilized as railway boosters. These are 60-kw., 125-volt, shunt-wound machines, but to meet the requirements of railway service a series field winding was added, consisting of 32 turns of No. 4/0 weatherproof wire, per pole; this winding was arranged with a single-pole double-throw switch to permit it being cut in and out of service. The four machines with a 300-kw. alternator, were belted to a 350-hp. engine, having a double-face flywheel, which carried the five belts, one running over the other.

To use the four machines as boosters for railway service, the series field winding is cut in and by referring to the diagram of connections, the booster circuit can be traced from the 550-volt busbar to the bottom throw of the switches in group *A* to the negative side of the booster, through the armatures and series fields (the switches of which are in the down positions) to the terminals 1, 2, 3 and 4 of group *A*. From here the circuit is carried to the corresponding terminals of switch group *B*, thence to the lines. When the switches of group *C* are in the up position the lines are fed direct from the 500-volt busbars, which is the case when no booster is required. When the switches of group *B* are in the down position the regular multipolar booster is in service.

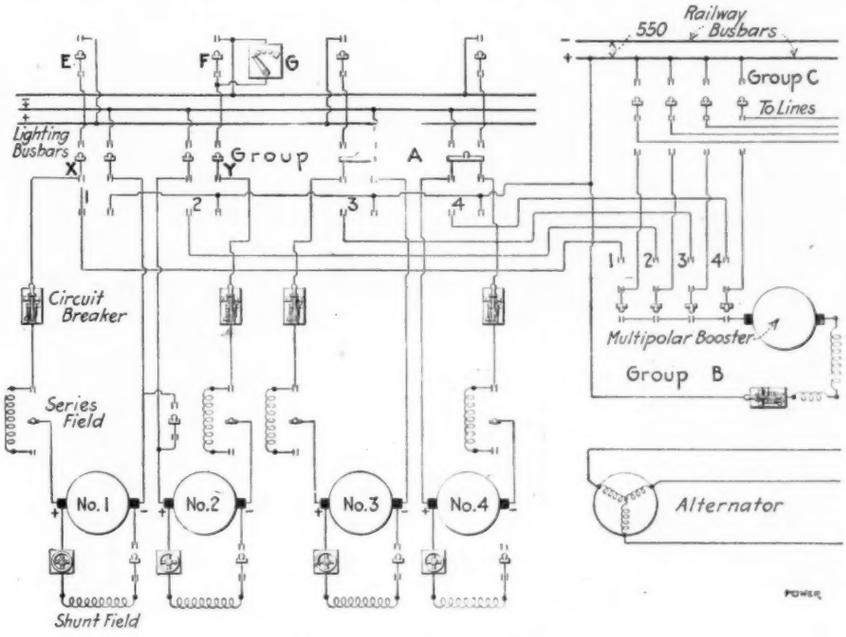


Diagram of Connections

To use the machines on the lighting system, which is an Edison 3-wire, 220-volt system, the series field is cut out by throwing its switch in the up posi-

tion, the shunt field switches are then closed and the machines are built up to the proper voltage. The switches of group *A* are then closed upward, connecting two machines in series. Next the switches *E* and *F* are closed, throwing the machines on the busbars. The same operation holds good for the other pair.

In summer it is customary to unhook the engine by taking off the connecting-rod and eccentrics, and drive the set by the alternator running as a synchronous motor. The set is started and run in the following manner:

Machines Nos. 1 and 2 are started as motors in series across the 220-volt busbars, the switches marked *X* and *Y* being closed in the up position, and the switch *D* is closed; this connects the machines in series. The shunt field resistance is all cut out and the positive switch *E* is closed. On the negative side a rheostat *G* is used in starting. It should be noted that these machines are belted to the alternator. The machines are brought up to speed (synchronous speed for the alternator) by the field rheostats and the alternator is then synchronized to the alternating-current busbars, not shown in the diagram. The alternator is now being run as a synchronous motor, driving the set. In winter the engine is connected again and drives the set.

In this way we have an emergency booster and lighting set besides a spare alternator for alternating-current purposes. This set has served for alternating-current lighting and railway boosting at the same time without any trouble, although not intended to do this.

CORRESPONDENCE

Repairing a Motor-Generator Set

In Fig. 1 is shown the plan view of a three-bearing motor-generator set, consisting of a three-phase induction motor directly connected to a direct-current generator. The motor has the bearings, one on each end frame, and the generator has one bearing at the commutator end. A short-circuit developed in the motor windings and the blaze that followed was fanned by the rotor about one-third of the circumference of the stator coils, damaging them beyond immediate repair.

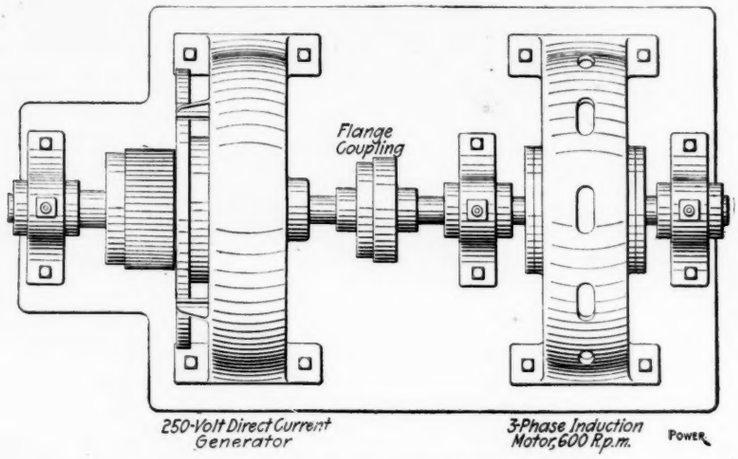


FIG. 1. PLAN OF MOTOR-GENERATOR SET

The problem was to repair the damaged coils and at the same time operate the generator. It was decided to drive the generator by a belt from a spare motor. Accordingly, the rotor of the motor with its shaft and coupling was removed, which left the armature of the generator unsupported at one end. As the rotor coupling had been shrunk on and was not easily removable, a new coupling and a short piece of shaft was made up, this shaft resting in the bearing of the motor.

The assembled shaft and coupling is shown in Fig. 2. The new coupling was slightly larger than necessary, but was counterbored to assist in centering with the old coupling. As only a limited supply of pulleys was at hand, and as the axial distance between the two machines was small, large pulleys of narrow face were necessary to get high enough belt speed to transmit the power.

The pulley was fitted as shown and after being bored out to suit the coupling only one pair of the original hub-clamping bolts remained. These were insufficient to hold or drive the pulley; therefore the driving was done by the arms indicated in the end view of Fig. 2. Extra holes were drilled through the couplings to take four bolts for the driving arms of 3x$\frac{5}{16}$-in. flat iron; the outer ends being fastened to the pulley rim by lag screws.

This pulley ran at a rim speed of over 5000 ft. per min. and carried a varying load up to 100 hp. This arrangement left the interior of the motor empty and permitted a workman to repair the damaged coils.

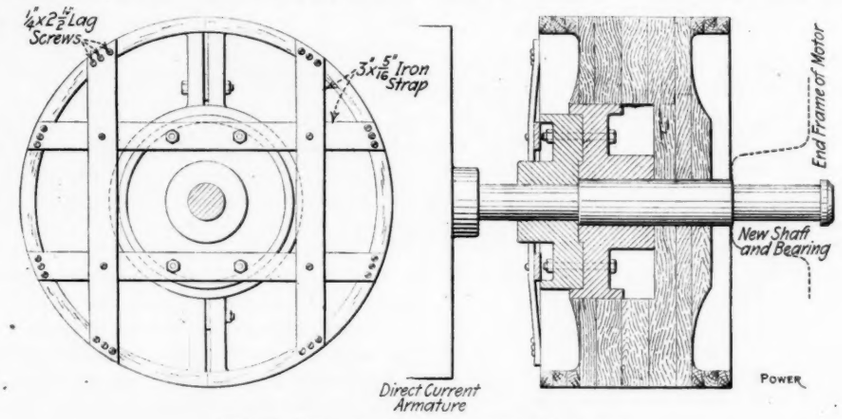


FIG. 2. METHOD OF ATTACHING COUPLING

GEORGE POWERS.

Canton, Ohio.

Generator Trouble

We have two 500-volt direct-current generators running in parallel, but which are about 4 miles apart; one is a 150-kw. and the other a 500-kw. machine. The larger generator frequently changes its polarity and causes trouble as the machines supply a trolley system. One side is grounded and where this machine changes over its positive side is connected to the rails. This does not appear to affect the load on the larger machine, but runs the load up on the smaller machine so that the circuit-breaker invariably goes out. We use sectional insulation with 6 in. of fiber or wood, and the current often jumps across.

When this happens we have to pull the section switches, a job that means loss of time as well as trouble. The 500-kw. generator is direct-connected to a large Corliss engine and the 150-kw. generator to a high-speed 240-hp. engine. I should be glad to receive suggestions from POWER readers as to where to look for the trouble.

W. F. LANDMESSER.

Herminie, Penn.

It is reported that a 150,000-volt transmission line is to be used in the Big Creek Development, a high head hydro-electric system which is being constructed in central California.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Some Details of the Cooper Gas Engine

Details, as a rule, play an important part in the successful operation of an engine, yet they are usually overlooked by the casual observer or are afforded but scant mention in technical descriptions.

During a recent visit to the works of the C. & G. Cooper Co., at Mt. Vernon, Ohio, the writer was very much interested in some of the details of their large twin tandem, double-acting, four-stroke-cycle gas engines. These are built in sizes from 200 hp. up, the one inspected being rated at 900 hp.

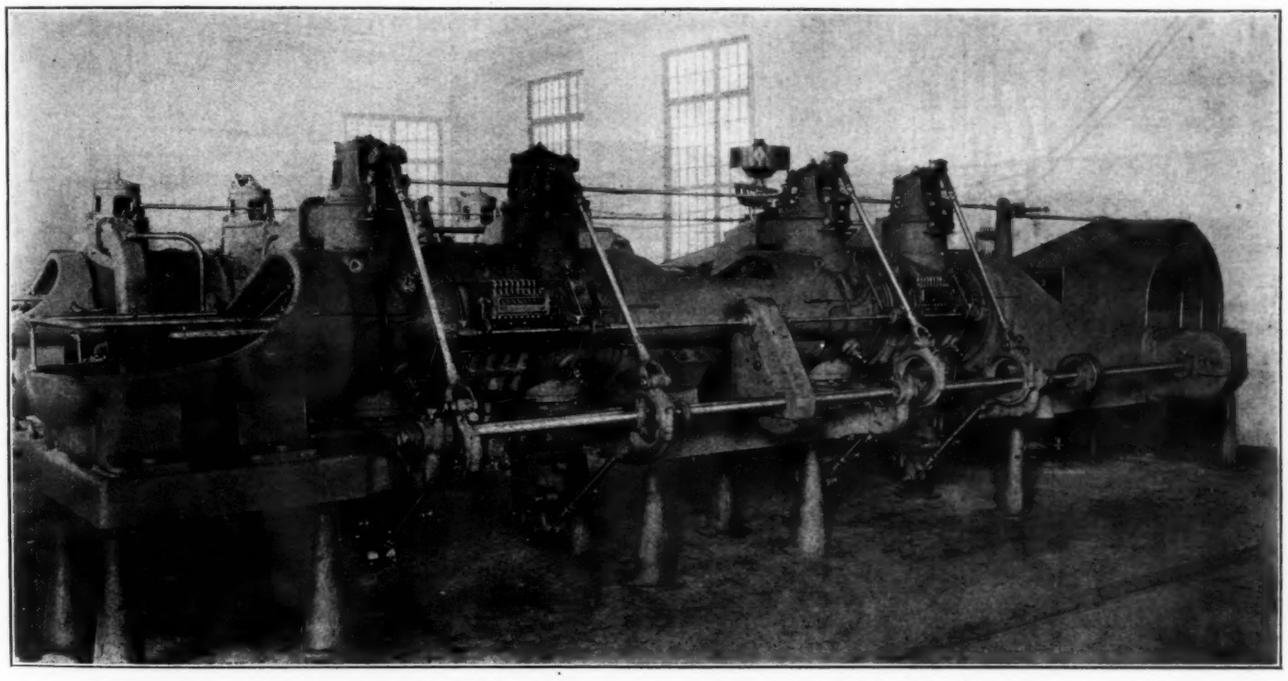


FIG. 1. TWIN TANDEM COOPER GAS ENGINE ON ERECTING FLOOR

Referring to Fig. 1, which shows one of these engines, one is first impressed with the symmetry of design. Considering the engine proper, everything is symmetrical about the middle and everything pertaining to one end of a cylinder is to be found at the other. This not only makes the parts interchangeable, but affords like gas and exhaust passages to the end of each cylinder insuring an equal charge to each. Furthermore, by driving the timing shaft from the middle the torsion effect is equalized.

INLET VALVE-GEAR

Of the various features, perhaps the inlet valve-gear is the most interesting. It is designed with the special object of affording uniform gas velocity and absolute control of the mixture at all loads. Referring to Fig. 2-a, which represents a section through the inlet-valve chamber, A is the valve held against the removable seat B by the spring C, the lower end of which presses against the split spider D; the latter also acts as the lower guide of the valve stem E. The upper part of the valve stem carries the guide nut F which is held at the top by the nut G. The bonnet cover H serves as the upper guide. A loose spring cap I, having a projecting collar is forced against the guide nut F by the spring, although sufficient play is provided between the flanges of this nut and the spring cap to permit the regulating valve to turn freely. A diametrical clearance on the collar avoids binding or sticking should the valve stem be out of alignment with the regulating valve.

One end of the multiplying lever K swings on the pin F' in the guide nut and the other is attached to the link L. A toe M is pivoted at one end to a lug H' on the bonnet cover and the other end is actuated by a pull rod driven through an eccentric from the layshaft. This is shown more clearly in Fig. 1. Hence, the inlet valve A and the regulating valve J to which it is rigidly attached open together against the action of the spring C.

Turning now to the means of regulating the mixture, Fig. 2-b shows a development of the sleeve N. This consists of two parts, an upper N' and a lower N''; the former is provided with fingers cut off at the bottom in steps and these fit into slots of unequal length between corresponding fingers on the lower section. In this way ports p' and p'' are formed for air and gas respectively. The lower part of the sleeve N'' is held fixed by dowel pins and the upper part is made adjustable vertically by a threaded ring Q, which is held in place by the lugs R. When this ring is turned by hand (see scale indicating width of port, Fig. 2-c), the ports p' and p'' can be proportional so as to give the proper mixture.

Surrounding this sleeve N are the air and gas chambers T and S, respectively, which have ports t and s corresponding with the ports in the sleeve.

Inside the sleeve the regulating valve J is provided with a double row of staggered ports j and j', shown in dotted lines at mid-position of the valve, Fig. 2-b. This regulating valve, besides reciprocating with the inlet valve A, also is made to rotate by means of the ring U and the arm W connected to the governor. This is ef-

fected by the ring having two blocks which slide in lugs on the valve; thus the valve *J* is allowed to slide up and down but is maintained in a fixed radial position determined by the governor and the desired amount of port opening is obtained.

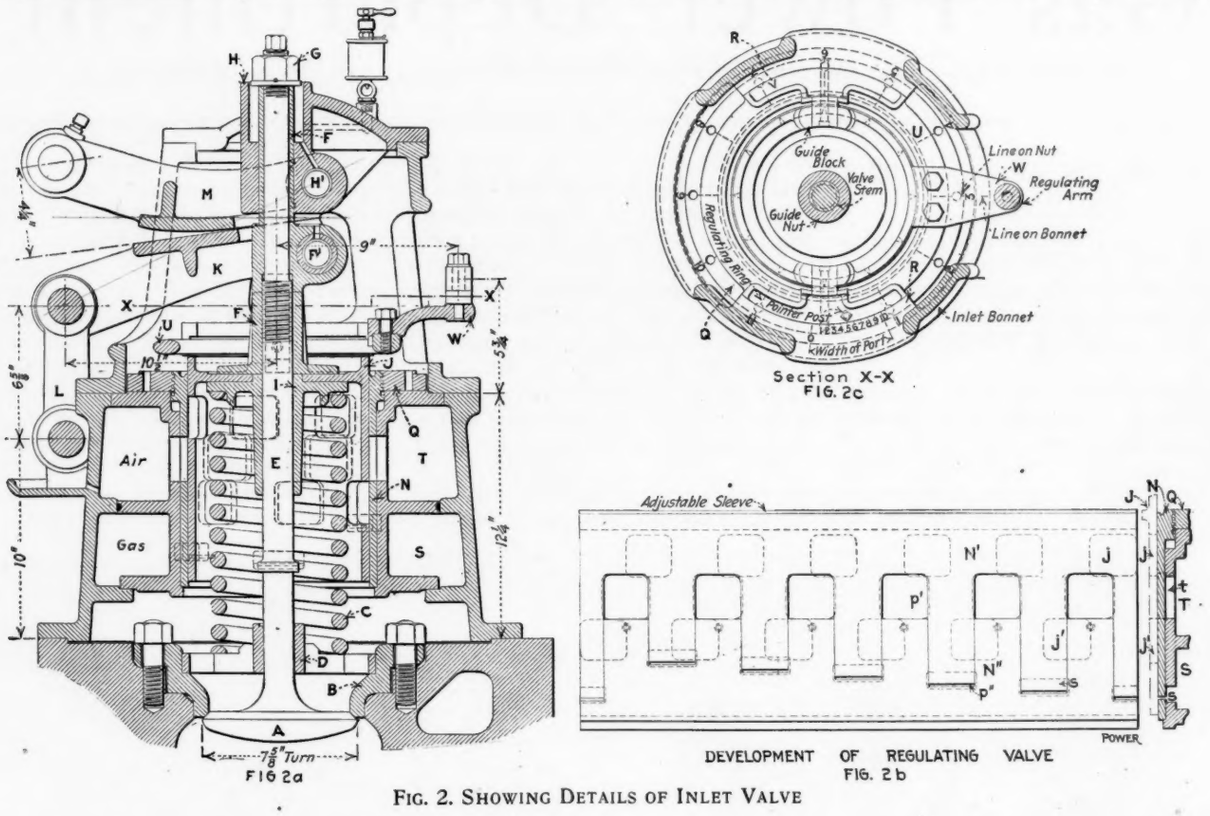


FIG. 2. SHOWING DETAILS OF INLET VALVE

AIR STARTING MECHANISM

As in practically all large gas engines, compressed air is employed for starting, their corresponding cylinders; hence, when the air is turned on all the valves but one automatically close and are opened in the proper order by the cam on the layshaft to admit air to the respective cylinders. In this way the incoming fuel mixture is not disturbed and the mechanism for this purpose being shown in Fig. 3. Mounted on the brackets which support the layshaft, are smaller brackets, each carrying a rod as shown. One end of this rod makes contact with the check valve in each line prevents the starting air from entering the cylinder after the charge has been ignited. After the air supply has been shut off the timing valves and rods fall out of action.

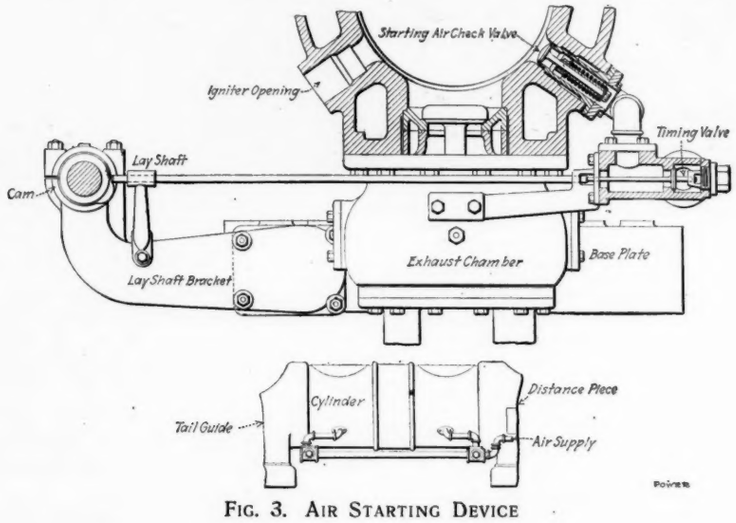


FIG. 3. AIR STARTING DEVICE

COOLING SYSTEM

Another feature worth mentioning in connection with the Cooper engine is the cooling-water system. As in most large gas engines, the pistons and piston rods are water cooled, but to prevent "sweating" of the rods, the water first passes through the cylinder heads and is then admitted to the rods and pistons. Thus it reaches the latter at a higher temperature. The intake into the rods is through a telescopic connection at the intermediate crosshead and the discharge, without reversal of flow, is at the main and tail guide crossheads.

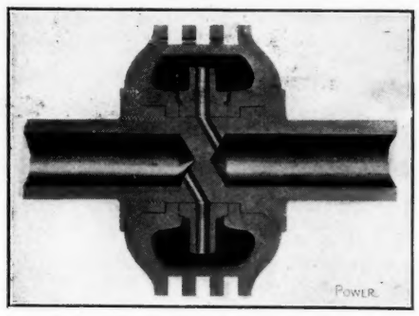
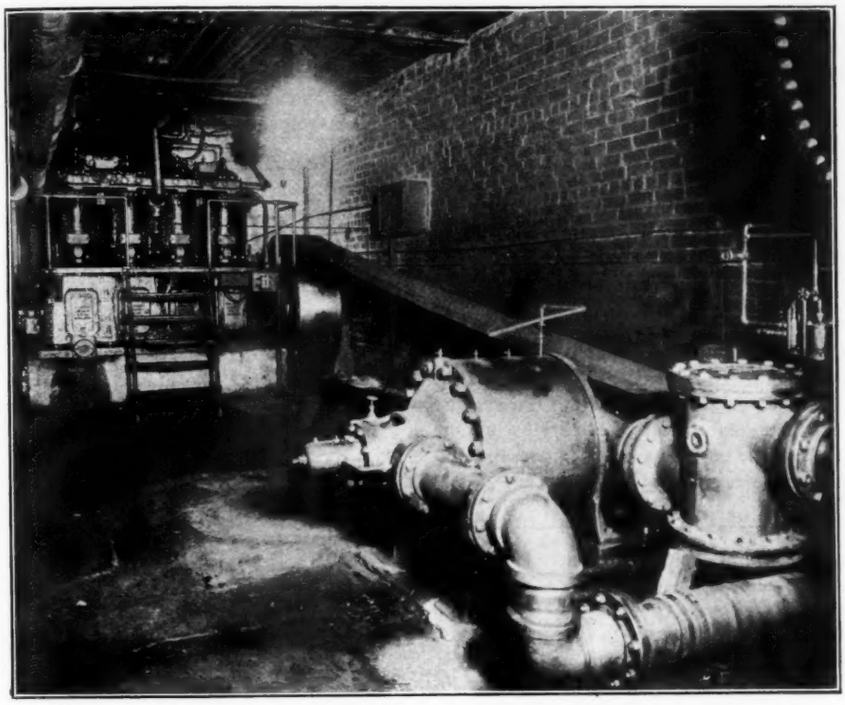


FIG. 4. SECTION THROUGH PISTON AND PISTON RODS

A section through one of the pistons and the rods is shown in Fig. 4, which also gives a clear idea of the method of attaching the pistons. The amount of circulating water required is from 6 to 10 gal. per b.hp.-hr., depending on the load.

Gas Engine for Elevator Service

The First National Bank building in Columbus, Ohio, contains a novel combination of centrifugal pumps driven by gas engines for hydraulic elevator service. The structure is a ten-story office building containing two hydraulic passenger elevators, each having a capacity of 5000 lb. and running at a speed of 500 ft. per minute. These are in almost constant operation during business hours and are in service from 6.30 a.m. to 9.30 p.m.



GAS ENGINE BELTED TO CENTRIFUGAL PUMP

As first installed, the elevators were served by one 9x12-in. triplex double-acting pump, driven by a 75-hp. motor and controlled by a Mason regulating valve which maintained a pressure of 140 lb. Power was purchased from the public-service company at 3c. per kw.-hr. and under these conditions the average cost of operation was $11.50 per day.

In addition to the high cost of operation, there was also serious objection to the noise of the pump gears, and it was decided to replace the electric motor with a gas engine and the triplex-plunger pump by a multi-stage centrifugal pump. The plant as so altered is herewith shown, and has been in constant service for 4 yr. at an average of 15 hr. per day.

The pump is a four-stage type built by the Alberger Condenser Co., and has a capacity of 600 gal. per min. against a head of 333 ft. A heavy multiple check valve eliminates any shock or hammer in closing.

The engine, built by the Bruce-Macbeth Co., is a four-cylinder, 100-hp. machine running at a normal speed of 250 r.p.m., and is controlled through a Mason bypass valve in combination with a special device on the engine-mixing valve. The action is such that when the pressure in the high-pressure tank rises above normal, the fuel and air supply is so reduced as to run the engine very slowly and at a small consumption of fuel. Natural gas is used for fuel.

The first cost of the gas engine and pump equipment was approximately $5000. The total operating cost under present conditions averages $62 per month, whereas with purchased current it was nearly $300 per month.

Preheating for Internal Combustion Engines

In its issue of July 28, *Elektrotechnik und Maschinenbau* publishes a review of an article contributed by A. Nougier to *Die Gasmotorentechnik* on the advantages of preheating the air in internal-combustion engines. The author says that with Diesel and similar engines the air must be compressed to from 425 to 500 lb. to reach a temperature above that of the ignition point of the fuel; in the case of crude petroleum, over 900 deg. F. This high compression is inconvenient as it necessitates increased weight of engine and requires a compressor for the air needed for fuel injection; it also presents further difficulties in maintaining tight pistons and valves. However, in the use of cheap fuels their high ignition temperatures must be reckoned with as these demand higher compression.

The method of securing a higher terminal temperature of compression by the injection of a small quantity of fuel shortly before the end of the stroke, its combustion so augmenting the temperature as to fire the later injected regular charge of fuel, has not proved justifiable, because the pressure still rose to over 560 lb. and the auxiliary fuel often failed to ignite. Also, the cylinder head was complicated by the two inlet valves.

An ample temperature rise can be attained without increasing the compression, by preheating the air drawn into the cylinder during the suction stroke whereby through a small initial heat increment the terminal temperature of compression is raised considerably even with a still lower terminal pressure. This preheating can be done by compelling the exhaust gases to pass through a form of heat reservoir over which the air passes on its way to the engine. Taking air at 60 deg. F. and atmospheric pressure it will be heated by the cylinder walls to say 100 deg. and through compression its temperature will be raised to about 1050 deg. This temperature, however, can be reached with a pressure of only 568 lb. if the air be heated to 176 deg., or with a pressure of 213 lb. by preheating to 227 deg.

In operating with the heavier oils or with tar oil it is necessary at first to run the engine with crude petroleum until the preheater becomes sufficiently warm.

The resultant lowering of the compression due to preheating of the air is of such advantage that the drop in thermal efficiency is negligible. On the other hand, the mechanical efficiency is considerably improved, and the process can at any time be applied to existent engines. It is not to be confounded with preheating of the air for the purpose of gasification, as of benzine, etc., nor with the lowering of the pressure at the end of compression by the injection of water.

Using crude petroleum, with a compression of 10 atmospheres and a terminal temperature of 1040 deg. F. the air would have to be preheated to 300 deg.

Such engines can very easily be built double-acting and working on the two-stroke cycle. The fuel control is simple and accurate and the average turning moment, and thus the duty for a determined speed or the speed for a given load, can be varied within sufficiently wide limits without changing gears. The carbureter and the ignition system disappear and in their place the air-injection arrangement, the fuel pump, and the preheater, the latter displacing the exhaust muffler, are all much less sensitive to derangement than are the parts eliminated.

Preheating, however, must not exceed a certain limit, as combustion would thus be impaired and other disadvantages might arise and it is therefore advisable to provide an air-temperature regulator.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Heating and Ventilating Northwestern University Buildings*

By J. M. Stannard

The heating and ventilating of the Northwestern University buildings, at Evanston, Ill., are accomplished by hot-water under forced circulation, and direct and indirect steam plants under both vacuum and gravity methods. The accompanying map, Fig. 1, shows the various buildings with the method of heating in each, the location of the power plant and the size and location of the feed mains.

The boiler plant consists of six boilers, four for the generation of steam and two for the reheating of the hot water. Two are horizontal return-tubular steam boilers 72 in. by 18 ft., and rated at 150 hp. The other two are Stirling water-tube boilers, each rated at 250 hp. The two for hot-water heating are horizontal return-tubular boilers, 66 in. by 16 ft., and rated at 125 hp. The water-tube boilers have Green traveling chain grates and the four fire-tube type are hand fired. During the more moderate cold weather only one of the hot-water boilers is fired for the reheating of the water, and two of the boilers are in use for generating steam. Besides furnishing steam to the radiation, the steam boilers supply circulating and vacuum pumps and one fan engine, for the induced-draft system.

For circulating hot water there are two Marsh single-cylinder double-acting circulating pumps, 10x12x12 in., each having a theoretical capacity of 5.87 gal. per single stroke. These pumps have revolution counters so that an accurate record of the water pumped may be kept. One large closed heater having 396 sq.ft. of heating surface made up of brass tubes, is used in addition to the two hand-fired boilers for reheating the circulation water. This heater was installed to utilize the exhaust steam from the various pumps and stoker and fan engines, so that no exhaust steam is wasted.

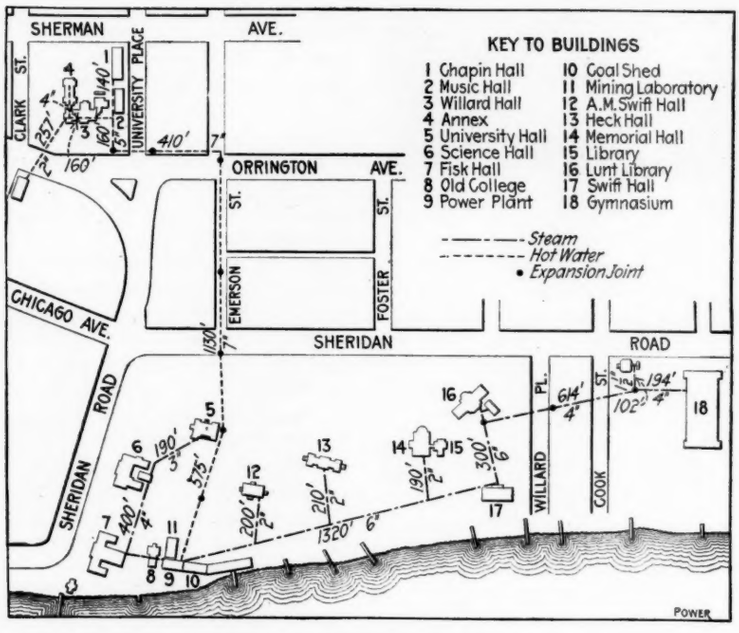


Fig. 1. Map of Buildings, Northwestern University, Evanston, Ill.

The hot-water system has a complete set of recording thermometers which show temperatures of outgoing and return water and outside temperature. In addition, regular thermometers are on the heater and the return line.

The steam-heating system consists principally of direct radiation supplemented with indirect in three of the larger buildings. The indirect radiation is for ventilating purposes, sufficient direct radiation being installed for heating only. Steam is generated at 80 lb. pressure and reduced in the power plant to 40 lb., for distribution to the 12 buildings. One to 5 lb. pressure is carried in the direct radiation and blast coils as weather conditions require, and is maintained by reducing valves in each individual building. Vacuum traps are applied to both direct and indirect radiation in two of the buildings.

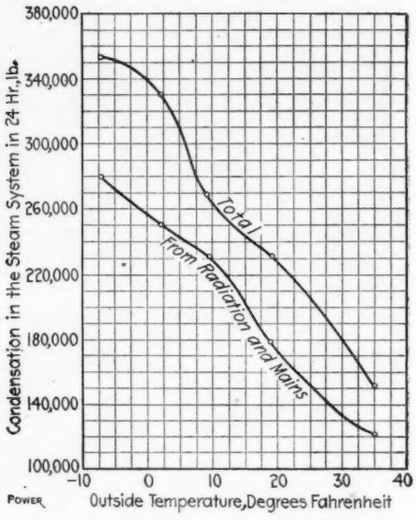


Fig. 2. Condensation in Steam System in 24 Hours

Three Marsh vacuum pumps, one 8x12x12 in., and two each 5x6½x10 in., are installed in the pump room and are used for holding vacuum on the steam-heating return lines and for handling the condensation from this system. The pumps maintain a vacuum averaging about 10 in., which drops to 2 to 3 in. where the returns leave the most distant buildings. The separate return lines are connected into two return mains which lead back to the power plant and are connected through settling chambers to the vacuum pumps in the usual way. The condensation is then delivered by the vacuum pumps to a receiving tank from

TABLE I. HOT-WATER TEMPERATURES FOR DIFFERENT OUTSIDE CONDITIONS

Outside	Ordinary		High Wind	
Temp. Deg.	Flow Deg.	Return Deg.	Flow Deg.	Return Deg.
60	115	95	125	100
50	120	100	130	105
40	130	105	140	110
35	135	110	145	115
30	140	115	150	120
25	145	120	155	125
20	152	125	162	130
15	158	130	168	135
10	165	132	175	137
5	173	135	183	140
0	181	140	191	145
−5	190	143	200	148
−10	200	150	210	155
−15	210	155		

*Abstract of paper read before the American Society of Heating and Ventilating Engineers, at Detroit.

TABLE 2. DATA SHEET, HEATING PLANT, NORTHWESTERN UNIVERSITY AT EVANSTON, ILL.

HOT WATER SYSTEM

Date from 10 a.m. to 10 a.m.	Average Outside Temp. Deg.F.	Pump		Temperatures				Heat												Cost 1000 sq. ft. Direct Rad. per 24 hr.
		Rev. of Circulating Pump per 24 hr.	Lb. of Water Pumped Calc.	Av. Temp. of Flow Deg.F.	Av. Temp. of Reheater Deg.F.	Av. Temp. of Return Deg.F.	Av. Temp. Added to Water by Reheater	Heat Added by Reheater B.t.u.	Total Condensation	Heat Added by Boilers B.t.u.	Total Heat Added to Water B.t.u.	Heat Deliv. to Reheater (Calc. 1000 B.t.u. per lb. cond.)	Heat Deliv. to Boilers (Calc. 11,000 B.t.u. per lb. coal)	Efficiency of Reheater Per cent.	Efficiency of Boilers Per cent.	Lb. Condensation from Reheater metered	Coal burned in H.W. Boilers No. 5 Coal at $2.35 per ton Lb.	Cost of Coal for Reheater	Cost of Coal for Hot Water Boilers	Total Cost for Heating Hot Water	
Jan. 5-6, 1912	—7	24,065	2,134,807	206	172	155	17	36,291,719	354,000	72,583,438	108,875,157	37,000,000	112,365,000	98	64.6	37,000	10,215	$8.46	$12.00	$20.46	$1.20
Jan. 8-9, 1912	2	25,306	2,162,549	206	180.4	157.2	23.2	50,171,136	330,500	55,361,254	105,532,390	52,000,000	91,520,000	96.4	60.5	52,000	8,320	$12.05	$9.78	$21.83	$1.27
Jan. 2-3, 1912	9	22,456	1,930,632	180.4	149	137.5	11.5	22,202,268	270,080	60,621,844	82,824,112	23,000,000	97,350,000	96.5	62.5	23,000	8,850	$4.99	$10.40	$15.39	$0.91
Jan. 29-30, 1912	19	21,383	1,857,389	178	151	135	16.	29,718,224	253,600	50,149,503	79,867,727	30,000,000	79,310,000	99	63.2	30,000	7,210	$7.18	$8.46	$15.64	$0.92
Dec. 22-23, 1911	35	19,073	1,649,661	144	126	117	9	14,846,949	132,400	29,693,898	44,540,847	15,000,000	52,030,000	99	57.1	15,000	4,730	$3.54	$5.56	$9.10	$0.53

which it flows by gravity to a feed-water heater and from there is taken by the boiler-feed pump and returned to the boilers.

The buildings on the forced circulation hot-water system are heated with direct radiation only. Temperatures and volume of circulation are regulated from the power plant according to Table 1.

The supply and return mains on the hot-water system are run in a special conduit, the bottom of which is made of book tile and covered with concrete. The pipes are covered with one thickness of high-pressure covering and a split sewer tile is cemented on the top. Expansion joints are placed at regular intervals. The expansion of water is taken care of by a tank in University Hall. The total hot-water radiation at present is 17,000 sq.ft.

TABLE 3. DATA SHEET, HEATING PLANT, NORTHWESTERN UNIVERSITY AT EVANSTON, ILL.

STEAM SYSTEM

Date from 10 a.m. to 10 a.m.	Average Outside Temp. Deg.F.	Condensation—Lb.							Coal Burned—Lb.					Evap. in Boilers per lb. Coal Calc.	Ave. Cond. per sq. ft. direct rad. per hour	Cost of Coal for Radiation and Mains	*Cost of Coal for 1000 sq. ft. direct rad. per 24 hr.
		From Radiation	From Hot Water Tank in Gym. Metered	From Reheater Metered	For Heating Swimming Pool Calc.	In Mains Calc.	For Heating Feed Water Calc.	Total Condensation	No. 2 Coal at $2.85 per ton Weighed	No. 5 Coal at $2.35 per ton Weighed	Pocahontas Screenings at $3.00 per ton Weighed	Total Coal Burned	Temp. Swimming Pool Raised Deg.F.				
		Metered	Metered	Metered	Calc.	Calc.	Calc.										
Jan. 5-6, 1912	—7	273,300	10,000	37,000	7,500	6,200	20,000	354,000	56,760	.....	.....	56,760	15	6.23	0.211	$63.93	$1.16
Jan. 8-9, 1912	2	245,600	2,000	32,000	6,000	6,200	18,700	330,500	53,795	.....	.....	53,795	12	6.14	0.19	$58.43	$1.06
Jan. 2-3, 1912	9	216,200	7,000	23,000	2,400	6,200	15,280	270,080	29,695	11,375	10,855	40,550	4	6.66	0.17	$48.20	$0.88
Jan. 29-30, 1912	19	172,200	9,000	30,000	3,000	6,200	13,200	233,600	29,250	.....	.....	40,625	6	5.7	0.14	$42.00	$0.76
Dec. 22-23, 1911	35	115,600	7,000	15,000	.....	6,200	8,000	152,400	9,410	19,195	.....	28,605	..	5.4	0.092	$28.72	$0.52

General Data.—Equivalent direct radiation in steam system, 55,080 sq.ft. Equivalent direct rad. in hot water system, 17,000 sq.ft.
*Includes condensation in mains.

For the steam system the distributing mains are: One 6-in. main for north buildings extending this size to between Swift Engineering Hall and Lunt Library through a 6-ft. concrete tunnel and 4 in. to Patton Gymnasium in a wood conduit. The total load on this north line is equivalent to 36,637 sq.ft. of direct radiation. In addition this line heats a hot-water tank for shower baths and the swimming pool, containing 66,000 gal. A 5-in. line in a brick tunnel carries steam to the south buildings, reducing at Science Hall to 3 in. for University Hall. The total load on this main is equivalent to 18,443 sq.ft. of direct radiation.

Branch lines are taken from these mains to individual buildings. The return lines are run parallel to the feed mains and are 3 in. in size at the power house. One of the interesting features

FIG. 3. HEAT IMPARTED TO HOT-WATER SYSTEM IN 24 HOURS

is the fact that the main return line at Patton Gymnasium is approximately 5 ft. below the high part of the tunnel near Swift Engineering Hall. The expansion of both supply and return mains is taken care of by slip expansion joints securely anchored at regular intervals.

Feed mains in the tunnel are covered with two thicknesses of 1 in. each of sponge felt covering. Return mains are covered with one thickness of low-pressure covering. Feed mains in the conduit are covered with one thickness of high-pressure covering, and packed together with uncovered return mains in oily shavings in a special wood boxing. The north supply main is dripped at Swift Engineering Hall, Lunt Library and Patton Gymnasium back into the return line through traps. The south main is sim-

ilarly dripped at Fisk and University halls.

All direct radiation in the buildings heated by steam is under the Johnson system of temperature control, and by-pass dampers in fan-ventilating systems are similarly controlled. For economical operation it was found that this temperature control must be installed.

Ventilation

Fisk Hall, Engineering Building and Patton Gymnasium have a fan system of ventilation, used intermittently as required. Science Hall, University Hall and Lunt Library have natural flue ventilation, and all other buildings have no provision for ventilation.

A system of records pertaining to the heating system is kept by the university. It was decided to select five days, the average temperatures of which would be as close as possible to the following: 10 below zero, zero, 10 above and 20 and 35 above. Readings and calculations cover coal burned, amount of condensation, temperatures, etc., for these days, all arranged on a comparative basis. The results are given in Tables 2 and 3.

All temperatures shown, except those pertaining to the swimming pool, are averages of 24 readings taken hourly between 10 a.m. and 10 a.m. The pump counters, meter and coal readings were taken once every 24 hr. at 10 a.m. Where no meter readings were to be had, as of the steam to the swimming pool, exhaust steam for heating feed water and the condensation in mains, the condensation to be charged to each was calculated as accurately as possible from the information at hand. The columns relating to heat, efficiencies and cost are calculated results based on actual records. In figuring the cost columns no account was taken of anything but coal burned. All curves are based on data taken directly from the figures and are submitted only to show graphically some of the more important items.

Steam vs. Hot Water Heating at Northwestern University

By Ira N. Evans*

The preceding article comparing the operating costs of hot water and steam heating at Northwestern University shows, on careful analysis, some curious facts. The different prices of fuel and boiler efficiencies so confuse the results that they are very misleading, when the economical generation of steam should have nothing to do with the relative costs of the two systems in operation.

The steam plant serves 55,080 sq.ft. of radiation at a maximum distance of 1620 ft. and comparatively little of the radiation at the end of the line. The hot-water plant serves 17,000 sq.ft. of radiation, most of it near the end of the line 2800 ft. from the power house. The maximum steam consumption of the steam system is approximately 14,000 lb. per hr., while that of the water system is about 4600 lb., or a little over one-third.

It is probable that the hot-water system was adopted on account of the extreme distance of the buildings from the power house and to utilize exhaust steam which would be wasted with a live-steam system operating at 40 lb. pressure.

The comparative distances and amount of insulated heating surface in the mains of the respective systems are given in Table 1. In round numbers there are 5500 sq.ft. of surface in the mains of the steam system, including the returns. The hot-water system has a total in mains of 9800 sq.ft. The steam system is operated at 40 lb. gage pressure (reduced from 80 lb.), or 286 deg. F., and assuming 1.8 B.t.u. per sq.ft. per hr. as the transmission and 40 deg. for the ground temperature the loss in radiation will be

$$(286 - 40) \times 1.8 \times 5500 = 2{,}435{,}400 \text{ } B.t.u. \text{ } per \text{ } hr.$$

TABLE I. HEATING SURFACE IN MAINS

Dia. Main in.	Length Main ft.	Sq. ft. surface Per lineal ft.	Radiating Surface sq.ft.
Steam System			
2	600	0.62	372
3	190	0.92	175
4	1200	1.178	1413
6	1620	1.734	2809
Total	3610		4769
Assume including returns 5500 sq.ft.			
Hot Water System			
2	800	0.62	496
4	320	1.178	377
5	320	1.456	466
7	4230	2.00	8460
Total	5670		9799

The steam on this system is all generated at 80 lb. gage and the returns come back at 10 in. of vacuum, or 192 deg. F., so that 1025 heat units will be available per pound.

Dividing by 1025 will give the pounds of steam

$$\frac{2{,}435{,}400}{1025} = 2376 \text{ } lb. \text{ } per \text{ } hr.$$

If the covering is 89 per cent. efficient the condensation will be

$$2376 \times 0.11 = 261 \text{ } lb. \text{ } per \text{ } hr.$$

which amounts to 6264 lb. per 24 hr. or very close to that calculated in the previous article. The loss in the hot-water mains, calculated in the same way with 180 deg. as the average temperature of the water, would be

$$(180 - 40) \times 1.8 \times 9800 = 2{,}469{,}600 \text{ } B.t.u. \text{ } per \text{ } hr.$$

Dividing by the 1025 B.t.u. to compare the loss with that of the steam system gives

$$\frac{2{,}469{,}600}{1025} = 2410 \text{ } lb. \text{ } per \text{ } hr.$$

and using the same efficiency for the covering the condensation is

$$2410 \times 0.11 = 265 \text{ } lb. \text{ } per \text{ } hr.$$

The condensation is practically the same as for the steam system, but while the loss by radiation on the steam mains is nearly constant throughout the season, that on the water system decreases with the water temperature in milder weather. Upon investigation it may be found that 6 per cent. of the steam used at different outside temperatures will measure very nearly the loss on the water mains. See Tables 5 to 8.

The great drop in temperature, 51 deg., and the low average temperature in the hot-water system tend to make it inefficient and unresponsive to quick regulation at the power house of the average temperature of the radiation. The 7-in. main is much too large and the great volume of water in the mains to be changed in temperature and its low velocity cause a considerable loss as the heat is not radiated when and where needed. The pumpage, 1500 lb., or about 180 gal. per min., gives a velocity of only 1½ ft. per sec. A 4-in. main could be utilized to better advantage, increasing the head to 1.52 ft. per 100 ft. of length or about 40 ft. on the system. The radiation surfaces of the water mains would be reduced by more than one-third and the loss in heat would be less by a like amount to say nothing of the decrease in first cost.

If the pumpage were doubled, decreasing the drop by one-half, the greater velocity in the radiation would mean higher efficiency and would allow lower average water temperatures to be used with the same heating surfaces, or if the same temperatures were maintained less radiation would be necessary to accomplish the same result. Doubling the pumpage would require a 5-in. instead of a 4-in. main. As the water is heated by live steam, this could be done without materially increasing the cost of pumpage over the present arrangement, as the extra power required to circulate the water is largely returned in heat due to friction.

TABLE 2. COST OF STEAM PER 1000 LB.

Cost Coal Per ton	Cost per 100 lb.	Evap. per lb.	Fuel per 1000 lb. Steam	Cost per 1000 lb. steam
$2.85	$0.1425	6.23	160.51	0.25873
2.85	0.1425	6.14	162.86	0.23207
2.89	0.1445	6.66	150.15	0.21697
2.71	0.1355	5.7	175.44	0.23772
2.515	0.1257	5.4	185.18	0.23277

The method of estimating the pumpage is not accurate as the slip and weight of water are not the same for different temperatures, and this in turn tends to

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give erroneous results when determining the total heat in the water.

Inasmuch as the distances and capacities of the two systems compared are so at variance, the loss in the mains should be separated from the comparison of the actual costs of heating. Various prices of coal are used with different evaporations, but as these factors have nothing to do with the relative efficiencies of the two systems, the most economical cost of generating the steam will be determined and used in the computations.

Table 2 gives the cost of steam per 1000 lb. and it is interesting to notice that the most expensive coal is the cheapest, costing $0.21697 per 1000 lb. of steam. Table 3 gives corresponding data on the hot-water boilers and shows that coal at $2.35 per ton gives better results than coal at $2.89 per ton on the steam boilers, the average cost being $0.1782 per 1000 lb. of steam.

The article states that the condensation is returned under 10 in. of vacuum, or 192 deg. F., by the pump. In all probability cold water has to be used for creating this vacuum, in which case some heat is wasted and the water overflowed if all condensation is returned to the plant. This is a charge against the steam system if the returns cannot be used for feed purposes. The steam for heating the feed water should be included in the steam-heating cost as presumably all exhaust steam is utilized and condensation may be returned at nearly 212 deg. F. from the water system.

All exhaust steam utilized in the reheater of the hot-water system will contain 970 B.t.u. instead of 1000 B.t.u. Except for a small loss by radiation from the heater shell, there is no loss as all the heat is either in the condensate or passed through the heater to the system. The efficiencies given for the reheater are probably accounted for in the differences in the actual and assumed values for latent heat.

In Table 5 the cost of the steam in the reheater is corrected so as to apply the price per 1000 lb. obtained with the steam boilers. The heat added by the reheater is given and dividing by 1025 will give the pounds of steam. The British thermal units in the coal used in the water boilers may be disregarded, as the weight and cost of coal is given and the number of heat units may be obtained from the records of the water system. The hot-water boilers show greater economy than the steam boilers, but their rated capacity is 250 hp., and the maximum load is 70 hp. and the minimum 36 hp., so it is not surprising, outside of the excessive estimate of water pumped.

The high-pressure boiler with a live-steam heater and gravity return for the condensation is decidedly more economical from an investment standpoint. The hot-water boiler is idle in summer and the fires are more difficult to handle economically for the wide ranges in outside temperature unless about three times the boiler power necessary is provided in the plant. Using one type of boiler for all purposes makes a more economical arrangement with a minimum of spare boiler power and a maximum overload capacity due to the interchangeability of the boilers. If this plant were properly arranged the entire load of both heating systems could be carried on the steam boilers and the hot water boilers eliminated or used in the steam-power plant as spare boilers. If the same generosity in boiler power were shown for the generation of steam as evidently has been installed for the water system the lower priced coal might be used in the steam boilers with a more economical showing.

Tables 4 and 5 give the costs per 1000 sq.ft. of radiation with and without the mains for both the steam and water systems, using the lowest cost given in Table 2 for generating steam. The data shows that the two systems are about on a par when the mains are omitted as the cost per 1000 sq.ft. of radiation on the steam system averages $0.8656 against $0.87 for the water. With the mains the cost on the steam system is $0.89 against $0.925 for the water, a difference of less than 4 per cent. This is as it should be with automatic heat control on the steam

TABLE 3. COST FOR WATER BOILERS ON SAME BASIS

B.t.u. from Water Boilers per 24 hr.	B.t.u. per hour	Lb. of Steam (1025 B.t.u. per lb.)	Coal per 24 hr.	Coal per hr.	Equiv. Evap.	Coal per 1000 lb. Steam	Cost @ $2.35 per ton
72,583,438	3,024,310	2951	10215	426	6.92	144.3	$0.1695
55,361,254	2,306,719	2250	8320	347	6.48	154.2	0.1812
60,621,844	2,525,910	2464	8850	369	6.68	149.7	0.1759
50,149,503	2,089,563	2039	7210	300	6.79	147.1	0.1728
29,693,898	1,237,246	1207	4730	197	6.13	163.2	0.1918
						Average	0.1782

TABLE 4. COST OF STEAM HEATING

Outside Temp. deg.F.	Lb. steam per 24 hr. feed water and radiation	Cost at $0.21697 per 1000 lb.	Cost per 1000 sq.ft. per 24 hr. without mains	Cost per 1000 sq.ft. per 24 hr. with mains
—7	293,300	$63.64	$1.1554	$1.1798
2	264,300	57.35	1.0412	1.0656
9	231,480	50.22	0.9117	0.9361
19	185,400	40.23	0.7304	0.7548
35	124,200	26.95	0.4893	0.5137
		Average	$0.8656	$0.8900

Mains 6200 lb. at $0.21697 per 1000 lb. = $1.34 per day = $0.0244 per 1000 sq.ft. of radiation.

TABLE 5. COST OF HOT-WATER HEATING

Outside Temp. deg.F.	B.t.u. from Reheater	Lb. steam (1025 B. t.u. per lb.)	Cost at $0.21697 per 1000 lb.	Cost Coal for H.W. Boilers at $2.35 per ton	Total Cost Heating with mains	Total Cost without mains (6% deducted)	Cost per 1000 sq.ft. with mains	Cost per 1000 sq.ft. without mains
—7	36,291,719	35,406	$7.68	$12.00	$19.68	$18.50	$1.158	$1.09
2	50,171,136	48,947	10.62	9.78	20.40	19.18	1.200	1.13
9	22,202,268	21,661	4.70	10.40	15.10	14.19	0.888	0.835
19	29,718,224	28,993	6.29	8.47	14.76	13.87	0.868	0.816
35	14,846,949	14,485	3.14	5.56	8.70	8.18	0.512	0.481
						Average	$0.925	$0.870

TABLE 6. TRANSMISSION ON STEAM SYSTEM

Outside Temp. deg.F.	Total Steam lb. per 24 hr.	B.t.u. per sq.ft. per 24 hr.	B.t.u. per sq.ft. per hr.	Average Temp. Steam-Room	B.t.u. per sq.ft. per hr. per deg. dif.	
					With Mains	Without Mains
—7	299,500	5574	232	215–70	1.60	1.57
2	270,500	5034	210	212–70	1.48	1.45
9	237,680	4423	184	212–70	1.30	1.24
19	191,600	3566	148	212–70	1.04	1.006
35	130,400	2427	101	212–70	0.71	0.677

TABLE 7. TRANSMISSION ON WATER SYSTEM

Outside Temp. Deg. F.	Total B.t.u. added to water per 24 hr.	B.t.u. per hour	B.t.u. per hr. without Mains (94%)	B.t.u. per sq.ft. per hr.		Average Temp. Water-Room	B.t.u. per sq.ft. per hr. per deg.dif.	
				With Mains	Without Mains		With Mains	Without Mains
—7	108,875,157	4,536,464	4,264,276	267	251	180.5–70	2.41	2.27
2	105,532,390	4,397,183	4,133,352	259	243	180.6–70	2.34	2.20
9	82,824,112	3,451,005	3,243,945	203	191	158.9–70	2.28	2.13
19	79,867,727	3,327,822	3,128,153	196	184	156.5–70	2.27	2.13
35	44,540,847	1,855,868	1,744,516	109	103	130.5–70	1.80	1.70

system, but the water system without heat control will be the cheaper to install. If the operating cost of the hot-water system were divided by the capacity in square feet of radiation which the 7-in. main could economically supply, or about 50,000 sq.ft. instead of 17,000, the cost would be considerably less than that of steam.

After all these costs have been deduced the water system uses more heat in proportion to the radiation than the steam system if the total radiation is given correctly in both cases.

In Tables 6 and 7 the total heat for both systems is given and the transmission rate per square foot of surface per degree per hour is determined under the temperature conditions given in the article. The data shows that the hot-water mains would have to be bare and added to the heating surface to transmit the amount of heat given at the rate for the steam system. At —7 deg. outside temperature all heating surface on the steam system should be in use and a rate of 1.6 B.t.u. only is shown, which is about right.

The water system with an average temperature of the medium about 30 deg. lower requires a rate of 2.2 to 2.4 B.t.u. to transmit the heat generated, which is impossible for the amount of surface given. If there are hot-water radiators that will give this transmission, it would be greatly to the credit of the hot-water system. These high values form a startling contrast to the efficiency of the steam radiators. The highest efficiency the writer has ever obtained is 1.8 B.t.u. with an average water temperature of 200 deg. and about 15 deg. drop on the job. Where the pumpage was furnishing one-half the heat in very moderate weather there was a seeming transmission of 2 B.t.u. per sq.ft., but this was due to the drop being made up by the friction of the water.

No doubt in moderate weather the low transmissions on the steam system are due to the heat control shutting off radiation, and this shows in the steam consumption. The steam job works out about right, but apparently something is wrong with the hot-water data; either the pumpage has been miscalculated or there is more heating surface. The article states that there is the equivalent of 36,888 or practically 37,000 sq.ft. of radiation on the steam system. The condensation is taken for the heating only and divided by 55 to obtain the cost per 1000 sq.ft. Is it possible that the capacity of the hot-water tanks and swimming pool is added to the system as radiation and only the steam consumption for the heating taken? If the cost of steam for the system were divided by 37 instead of 55 it would mean an increase per 1000 sq.ft. over the quantity given of 50 per cent., and would make the cost of steam heating relative to hot water much in favor of the latter.

Temperature Equivalents of Wind Velocities*

By W. H. Whitten

A comparison of records taken at the group of buildings of the Harvard Medical School on the total heat expended and average temperatures and average wind velocities showed that 1 mile of wind movement per hour required substantially the same amount of heat supply as 1 deg. change in temperature. A further study of similar records, however, has shown that there is a greater proportion of loss due to wind movement as the temperature drops.

This led the author to make investigations as to the impact effect of wind of the same velocity at different temperatures. It was found that there is a regular rate of increase in effective pressure as the temperature drops, although the wind velocity remains constant. This regular rate of increase of pressure is maintained only while the barometer readings are normal. A barometrical change caused changes in the impact pressure. The exact rate of this change was not determined, but the fact of such change was detected, the tendency being for an increase in pressure as the barometer rose and for a decrease as it fell.

It was estimated that, with the barometer and the wind constant, the increase in pressure is 0.4 per cent. per degree drop in temperature. It was also found that the nonpressure or suction on leeward sides of buildings increased in about the same proportion. The point at which heat loss from 1 mile of wind movement per hour and temperature were equal seemed to be between 36 and 39 deg. above zero. Above this temperature, the effect of wind became less important than the temperature changes, and below it, correspondingly more important.

For example: If, at 37 deg. plus, 1 mile of wind movement per hour is equal to 1 deg. drop in temperature, at zero 1 mile of wind movement per hour will equal

$$1 + (37 \times 0.004) = 1.148 \textit{ deg.}$$

If the temperature increases to 50 deg. plus, then 1 mile of wind movement equals

$$1 - (13 \times 0.004) = 0.948 \textit{ deg.}$$

As a rule for personal guidance, the author has adopted the following: From 40 to 15 deg. plus, 1 mile of wind movement per hour is equal to 1 deg. drop in temperature; from 15 deg. plus to 20 deg. minus, 1 mile of wind movement per hour is equal to 1.15 deg. drop in temperature. This is for buildings constructed in the ordinary manner, that is, without protected windows. Applied strictly to the glass surface, with leakage standardized, the loss from wind movement may be calculated as only three-sevenths of the loss under usual and ordinary conditions. This not only applies to the sides having the so-called greatest exposure, but, owing to the suction or nonpressure existing on the sheltered sides, should be applied to all sides of the building.

*Abstract of paper read before the American Society of Heating and Ventilating Engineers, at Detroit.

Location of Ventilating Fans

The improper placing of circulating fans often lowers the efficiency. Forcing or drawing the air around short turns means a waste of power or less air movement for the same amount of power applied to the fan. Fig. 1 shows an exhaust fan placed against the wall at *E*

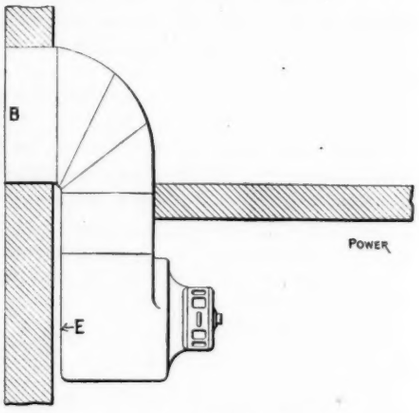

Fig. 1. Fan below Outlet

for removing hot air through the opening *B*. Originally the flue had a square corner, but with the long ell shown in the drawing the fan works more efficiently.

A better location for the fan is shown in Fig. 2 at *C;* the propeller is of the inclosed type directly connected to the motor. Air is pulled up through the opening *X* and forced out at *B*. The friction is reduced and the air is taken from

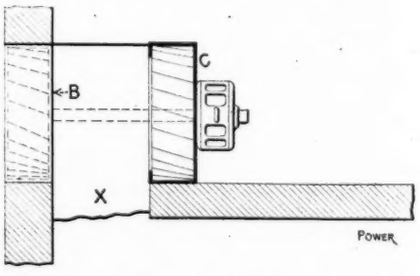

Fig. 2. Better Location for Fan

the hottest part of the room. The propeller could be set at the point *B* in Fig. 2 with the shaft carried back to the motor, as indicated by the dotted lines and better results obtained than with the installation illustrated in Fig. 1.

C. R. McGahey.

Baltimore, Md.

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POWER

Preparing for License Examinations

It is natural for the novice in steam engineering to be imbued with ambition and it is usual for him to take the shortest route to advancement. It is the license he is after with very little thought of anything farther. The advice often given the novice by those who have secured a license is sometimes misleading and frequently responsible for failure. Some advise the study of questions and answers in catechism form; others will give him questions, which the examiner of the district is in the habit of asking, and the answers which are thought to be satisfactory. Either of these "short cut" methods, while helpful in a way, is likely to be productive of disappointment, for the examiner as a rule will readily discern that the answers to his questions are "stocked," and a little divergence from his usual manner of questioning may cause confusion to the applicant.

In these columns a correspondent recently suggested changing around from plant to plant so that familiarity with various types of equipment might be had as an aid to securing a license. This is well enough as a part of the young man's training, but familiarity with the use of steam-plant equipment seldom teaches the principles of its action. It is this knowledge that the student of steam engineering should endeavor to possess, and it can be attained only by systematic and persistent study of the fundamental principles and the natural laws which affect the generation of power and its transmission.

Presuming that the beginner has a rudimentary education in common school studies, the logical procedure is to take up a course of study, beginning with physics, principles of mechanics, machine elements, and the kindred subjects to which his studies will progressively lead. The mere memorization of facts and theories, however valuable when properly used, should by no means be the main object of this study. There should be a constant aim to clearly understand what is read, to develop the faculties of observation and reasoning, and to rightly use what is learned. Only by using and adding to it can the student make it his own.

When sufficiently advanced in these studies, and with a reasonable experience in actual steam-plant work, the applicant for license will be able to go before the examiner and discuss the subjects on which the examination questions are based, giving answers that show an intelligent understanding of the principles involved, which invariably will impress the examiner favorably. The value of a thorough educational foundation does not end in the examination room. It will be found of far more value in the practical operation of the plant. For the problems which must be solved in actual work are readily mastered when the basic laws are understood.

The foregoing is neither new nor original, but is the basis of education as taught in colleges and technical schools, and is especially directed to those who, without preparation, have secured employment at the trade, and find it difficult to advance because they study at random and with uncertain result, when the same effort properly directed would lead to success.

Getting the News

Gratefully we acknowledge our appreciation of the interest shown by our readers in sending us newspaper clippings of boiler explosions, wrecks, flywheel accidents and other matters pertinent to POWER's columns.

But— (and this "but" looms large in this instance) early newspaper reports of such happenings are notoriously inaccurate, and necessarily so when one considers that they must be published a few hours after they occur. It is manifestly unfair to lay much blame upon the average reporter for his inaccuracies as his lack of technical training and mechanical unfitness excuse his not detailing the causes of these accidents.

Whenever it is possible, we ask that our readers who are located near the scene of the occurrence, and their occupation allows it, will endeavor to ascertain all the facts, the cause, how it might have been avoided, or at least try to verify the daily-press report. If you cannot get this news, ask someone who can. A photograph often helps to make the story clear.

As stated in our foreword in the issue of Mar. 12, "Don't let anything stop you. Be the *first* to send the news!" Remember, there are over thirty-one thousand readers in the POWER family, and we ask that each one make extra efforts to reflect additional credit on the family. We will bear all the necessary expense attendant upon securing all possible information of interest to our readers, and we will pay you the usual space rates besides.

Incomplete Data

A statement often seen in engineering articles is to the effect that the subject mentioned will necessitate more space than the author cares to give or thinks it deserves.

Unfortunately this frequently renders the article of much less value than if the author had covered a single subject thoroughly, rather than attempt to cover several subjects in the same space.

For instance, a book recently published contains less than ninety pages, yet the subjects it embraces include boilers, steam engines, turbines, and oil and other types of internal-combustion engines. Any one of these subjects deserves several volumes to adequately treat of it. As a result the volume is practically valueless to the purchaser, and is a disappointment in every way.

The idea is this. Suppose one is writing an article dealing with furnace combustion. A dozen pages or more may be devoted to giving directions for stopping up cracks in the setting, how to fire properly and control the draft, but if the author stops there the real meat of the article is missing. Engineers want to know the why of things; why air leaking into the furnace will adversely affect the economical combustion of the fuel; why it is not just as practicable to fire heavy as light, and what the draft has to do with the process of combustion.

Telling the reader that these conditions will produce a low CO_2 and then dropping that phase of the question, because the space available is insufficient, is disappointing. If the reader requires information as to how to handle his fire, he will also require information regarding flue gas and why faulty conditions affect a high percentage of CO_2.

It is one thing to know such and such is a fact, but it is more important to know why it is so. Stating facts by halves is much like eating partly cooked food, most unsatisfactory. Both should be given longer treatment and done to a turn.

Discussion vs. Comment

Our Sherlock Holmes thinks he has detected a disposition on the part of certain contributors to take the slightest pretext that any article affords to send us some comment to print. In his dyspeptic moments, he has gone so far as to declare that some evidently search the paper each week for something on which to hang an excuse for writing a letter, which, as a rule, simply reiterates what has already been said, or calls attention to the most obvious conclusions that anyone could reach from a mere reading of the original matter.

We sincerely hope that he is wrong in this supposition.

Discussion is wanted—not mere comment, and perhaps it may not be amiss for us to make a little explanation of just what we are after.

POWER aims to help the reader in every possible way and wants all the good practical information obtainable from those constantly meeting perplexing problems and solving them in ways that would interest others.

Not for a moment would we discourage anyone from calling attention to anything that may be wrong as described by another or that might be done in a better way. Such are particularly desirable. Whenever an article or a letter recalls to a reader an experience that throws a new light on a subject, we do indeed want to hear of it, but rambling comments that deal only in generalities and get nowhere with conclusions are no use.

Give the matter a little thought and you will not go wrong. When you write us, write not merely for the sake of writing, but because you have something really good to offer. In short, use POWER as it is meant to be used.

Being Contented

Now and again some well intentioned correspondent opens his tear-ducts and floods our editorial sanctum with a stream of grievances against his employer or chief. The more trivial the cause, the more inclined is he to hope that "some brother engineer" will take up the cudgels and provoke a long and useless discussion. Perhaps we should acknowledge our own culpability in that we have permitted too free discussion of grievances. If so, we have erred in our leniency.

Next to war and pestilence, the individual most to be shunned is he who is continually complaining. He dampens the ardor of his associates; he works unwillingly and halfheartedly and, sooner or later, he is let go as inefficient or otherwise undesirable.

A powerful factor in the betterment of almost any man's condition is contentment—at the least, patience. But as a celebrated divine has aptly said, when a man is perfectly satisfied with himself it is time for him to die. It is self-dissatisfaction that a man should cultivate; he will then have a clear insight in placing blame where it properly belongs. In a few words, contentment is born of self-dissatisfaction.

"If you do not like your boss, get a new one" is usually much easier said than done, but a rigid self-examination frequently reveals the fact that it is not always the other fellow in whom all the faults lie. The self-satisfied individual in his warped judgment makes mountains out of molehills, and scents a reproach in every order and a criticism in every look. The wise man, dissatisfied with himself, finds his real satisfaction in self-study.

Let us first to ourselves be true. If we discover that the mote is lodged in our own eye, our mental vision will not then distort the actions of those about us, and instead of grumbling and whining over adverse conditions, we will seek to ameliorate them. No one loves a "kicker." Let us cultivate contentment as a most valued possession. Then the daily task will be less difficult, we will more patiently bear with others, and the compensation will be satisfaction in our well-doing.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Cemented and Sewed Belt Joint

Much has been written pro and con on the subject of belt splices, but I have a very good splice that has not been mentioned and will describe it for the benefit of others. It takes more time to make the splice, but I consider it better than either wire lace or cement alone and it will last as long as the belt. The ends are scarfed and cemented with glue in the usual manner and then sewed with ¼-in. rawhide lace leather, as shown in Fig. 1. To do this a two-edged knife is used. It is seen from Fig. 1, that the holes for the stitches are cut at an angle to the side of the belt. The seams are placed about 2 in. apart and the lace is pulled in tight, the stitches being about

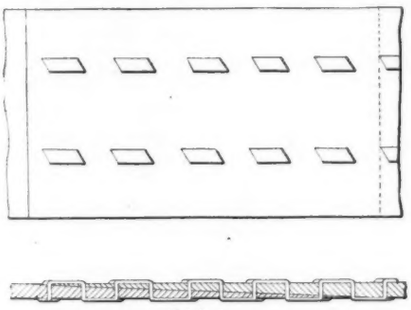


FIG. 1

CEMENTED AND SEWED BELT JOINT

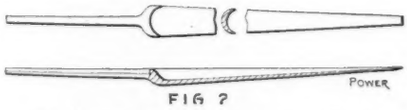


FIG. 2

TOOL FOR PUNCHING LACE HOLES

½ in. long. This makes a smooth, pliable joint, and one that cannot pull apart.

Of the numerous belt fasteners on the market nearly all make a stiff joint and will break the belt at the end of the fastener, as the belt is cut nearly one-third off in driving the fastener through, especially when used on small high-speed pulleys.

In making a splice with rawhide lace I begin in the center of the belt and work to the ends of the joint and back to the center. To make the lace holes I had a cutter made as in Fig. 2. This is forged from a piece of steel, like a half-round file hollowed out and filed to a sharp cutting edge. It is easy to push the sharp point through the belt and give it a half turn. If a larger hole is required, the cutter is pushed down and turned until the hole is the right size. It is better to bore from both sides because in a thick belt the hole will be tapered if bored from one side only.

In making a cement splice, I use a block plane to scarf the ends and common liquid glue as cement. I have a 2½-in. single-ply leather belt running a brine pump in a wet place. This belt has a 4-in. splice made with liquid glue, not sewed, that has been in constant use nearly one year without the splice starting to open. There is also a glued-and-sewed governor belt that has been in use over four years without any damage to the splice.

The mechanic that informed me about the sewed splice said that he had made several such in 36-in. belts, which gave good results and in which the rows of stitches were 1 in. from the edge and 2½ in. apart.

J. C. HAWKINS.

Hyattsville, Md.

Diagrams from Water End of Large Pump

The diagrams shown were taken from the low-pressure, intermediate- and high-pressure pump chambers of a vertical triple-expansion, pumping engine, 28 and 54 and 80 by 60 in., with 30-in. plungers and of 20,000,000 gal. capacity. The terms low-pressure, intermediate- and high-pressure are used to designate the pump chambers under the low-pressure, intermediate- and high-pressure cylinders, respectively.

The inlet and discharge pressures, as will be noticed, are practically the same on all three pumps. The pump receives water from a 48-in. supply pipe leading from a reservoir. The level of water in the reservoir is maintained at a height sufficient for water to flow into and fill the pump, as will be noticed by the atmospheric line, which runs parallel with the inlet line of the diagram, except for a slightly higher pressure in Fig. 1, from the low-pressure pump. This is due to the fact that the pump is set so that the low-pressure pump is nearest the supply pipe.

It will be interesting to follow the lines of the diagrams in the direction of the arrows and note the variation of pressure and the causes of the irregular lines. Starting at *A*, on the low-pressure diagram, Fig. 1, it will be noticed that as the plunger descends, the pressure rises considerably higher than that in the mains. This is due to the inertia of the valves and the resistance of the outlet caused by a heavy spring, intended to assist the valve in seating quickly.

The diagrams were taken when the pump was running at a discharge rate of 22,000,000 gal. per 24 hr., which gives a plunger speed of 270 ft. per min. With a 60-lb. spring it is found that the pressure raised to 115 lb., at point *B*. The pressure in the discharge line being 85 lb.

It will be observed that after valves

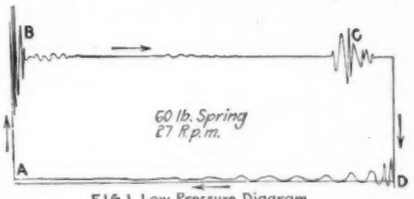


FIG. 1 Low Pressure Diagram

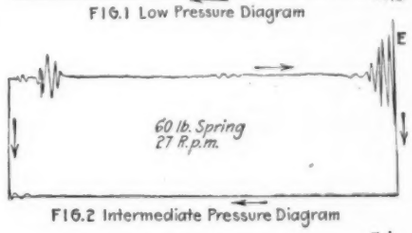


FIG. 2 Intermediate Pressure Diagram

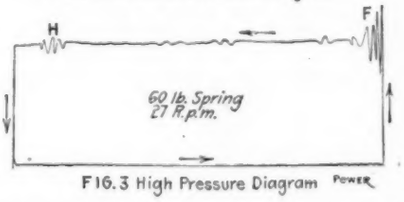


FIG. 3 High Pressure Diagram POWER

DIAGRAMS FROM WATER END OF LARGE PUMP

are fully opened the plunger displaces its volume without any variation in pressure until *C* is reached. The fluctuation at this point occurs when the plunger of the low-pressure pump has almost reached the end of its stroke and, so far as its displacement is concerned, there is no reason why there should be any variation until the plunger had reached the end of its stroke and started to rise.

By carefully watching the action of the pump we find that the intermediate plunger starts descending at this time, and the irregular line at *C*, Fig. 1, and at *E*, Fig. 2, are caused simultaneously, for when the low-pressure plunger is nearing the end of its stroke, the intermediate plunger starts descending, thereby adding volume and pressure to the discharge line. Fig. 3 is from the high-pressure pump chambers, the plunger commencing its downward stroke at *F*

and discharge takes place without any variation until the point *H* is reached, when the displacement of the low-pressure plunger causes a slight irregularity in the pressure line. As the low-pressure pump is a considerable distance from the high pressure, the effect is not so noticeable.

A singular thing is that when the plunger starts descending, and as the valves are forced slightly open, evidently at the instant when the pressure in the pump chamber is overcoming the pressure in the discharge line, water in the latter momentarily flows back to the pump chamber. I believe the irregular lines at the commencement of the plunger stroke prove this.

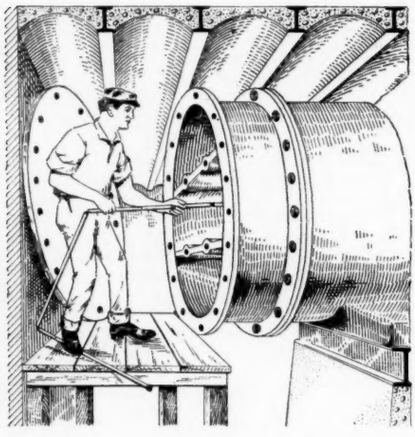
JOHN BURNES.

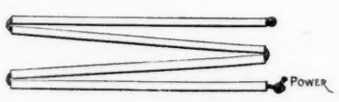
Columbus, Ohio.

Condenser Tube Cleaning Kink

A kink useful to many engineers in small plants where space is limited, is herewith shown.

The tubes on the intake side of the condenser which we have in service became plugged with small stones, twigs and mud, and how best to clean them puzzled us for a time. The head was taken off, and at first we tried to wash the tubes out, but found this of no avail. As we had but 4 ft. between the condenser and the wall, using a straight rod was out of the question, so we hit on the plan shown.





APPLICATION OF JOINTED CONDENSER TUBE CLEANER

A length of ¼-in. pipe was cut into 3-ft. lengths, a couple of strands from a steel-wire rope were passed through and made fast at one end by soldering, and a knot tied in the other end, leaving it a couple of inches longer than the pipe to allow for bending. This made a good form of jointed rod, easily handled in the close quarters, and good time was made in cleaning out all the tubes.

ARTHUR H. HODGES.

Hudson, N. Y.

Sand Submerged Suction Disables Condenser

While serving as troubleman for a large company, I was sent to a town to locate and remedy the trouble which a water-works concern was having with an independent jet condenser which my company had furnished. I was met at the station by the owner of the plant, a judge of two counties, by the way, and escorted post haste, over to the pumping station where the disabled condenser was located. I obtained a brief outline of the situation, while looking matters over, as to the behavior of the condenser and tried operating it myself; its action corroborated the engineer's remarks.

I concluded that the trouble lay in the suction pipe, so after a description of the layout of this line from the suction well, which was covered with about 10 ft. of snow, to the condenser, I took off the condensing cone and discovered a large stone about as large as a medium-sized cantaloupe lodged in the neck. This immediately aroused my suspicions and I made further inquiries about the suction well, being assured by owner and engineer, emphatically, that it was clean and clear and that the suction pipe was an ample distance above the bottom. I arranged to have one of the laborers clear the snow away from the top of the well while I made a still further examination of the interior of the condenser, finding everything in good condition.

I then went down to the well, which was uncovered by this time, and with the aid of a long pole on the end of which was a hook, discovered the end of the suction pipe to be submerged in sand to the extent of about 6 in. This sand the river had deposited during high-water periods.

The judge was with me when I made this discovery and it required no jury to help him form his opinion, as he immediately sentenced one of his men, the one who had been most positive that the well was clear, to put on hip boots and clear out the sand. After this was done, I started the condenser and raised and maintained the required vacuum without any trouble.

W. OLIVER.

Racine, Wis.

Suitable Form of Packing List

This copy of our packing list, and the manner of its arrangement, may prove useful to engineers in plants where there are many machines which require many sizes of packing.

In making up a list of packing and the machines in which it is used, it is usually best to arrange it as shown. The name and kind of packing should head the list of machines in which it is used. Then the names of the various machines and the location of their stuffing-boxes should follow. Opposite each stuffing-box item should be stated the size packing it requires.

PACKING LIST

Plant of .....

"A's" Steam Ring Packing

Engine No. 2	
Piston rod h.p.-cylinder	1 11/16 x2 3/4 in.
Piston rod l.p.-cylinder	2 3/4 x4 in.
Valve rod h.p.-cylinder	3/4 x1 1/2 in.
Valve rod l.p.-cylinder	1 1/8 x1 1/2 in.
Engine No. 3	
Piston rod	1 3/4 x2 1/4 in.
Valve rod	3/4 x1 1/2 in.
Engine No. 4	
Piston rod	1 3/4 x2 7/8 in.
Valve rod	1/2 x1 1/2 in.
Fan engine	
Piston rod	1 13/16 x1 7/16 in.
Valve rod	5/8 x 1 1/16 in.
Fire pump	
Piston rod (both ends)	1 3/4 x2 3/4 in.
Valve rod	1/2 x1 1/2 in.
House pump	
Piston rod (both ends)	1 5/16 x1 7/8 in.
High pressure water pump	
Piston rod	2 1/2 x3 1/8 in.

Boiler feed pump	
Piston rod (steam end)	1 1/8 x2 in.
Valve rod	3/4 x1 in.
Drinking water pump	
Piston rod (both ends)	7/16 x1 3/4 in.
Valve rod	1/16 x 1 1/16 in.
No. 1 Vacuum pump	
Piston rod (both ends)	1 1/2 x2 1/4 in.
Valve rod	5/8 x1 1/16 in.
No. 2 Vacuum pump	
Piston rod (both ends)	7/8 x1 3/4 in.
Air compressor	
Piston rod (both ends)	1 1/4 x1 1/4 in.
Brine pump	
Piston rod (both ends)	1 1/2 x2 3/4 in.
Valve rod	3/4 x1 1/2 in.
Ammonia pump	
Piston rod (steam end)	1 3/8 x2 1/4 in.
Valve rod	3/4 x 1 1/16 in.

"B's" metallic packing

Engine No. 4	
Piston rod	1 3/4 x2 7/8 in.
"C's" metallic packing	
Engine No. 1	
Piston rod	3 x4 5/8 in.
Valve rod	1 11/16 x2 5/8 in.
Expansion rings for above packing	3 x4 1/16 in.
Hemp packing	
Ammonia pump (ammonia end)	5/8 in.
Boiler feed pump (water end)	1/2 in.

"A's" Ammonia packing-spiral

Ammonia pump	
Piston rod (ammonia end)	1 3/8 x2 5/8 in.
Boiler feed pump	
Piston rod (water end)	4 x5 in.
"B's" spiral	1/4 in. 3/16 in. 1/8 in. 1/16 in.
"A's" spiral	5/8 in. 1/2 in. 1/4 in.
"A's" spiral-ammonia	5/16 in. 9/32 in. 1/4 in. 3/32 in. 3/16 in. 3/32 in.

The list should be framed behind glass and hung in a place convenient to where the packing is kept. Another way is to paste the list on a board of the proper size and cover thoroughly with shellac, but the glass-paneled frame is preferable.

The date of receipt of each size and style of packing should be kept, also the date and amount used in each machine. This enables one to determine how much packing any particular machine uses in a given time, and also reveals the fitness of the packing for the work required of it.

It pays to keep packing in separated compartments, each labeled, so that a glance at the label will indicate the size of packing contained in the place or places in which it is used.

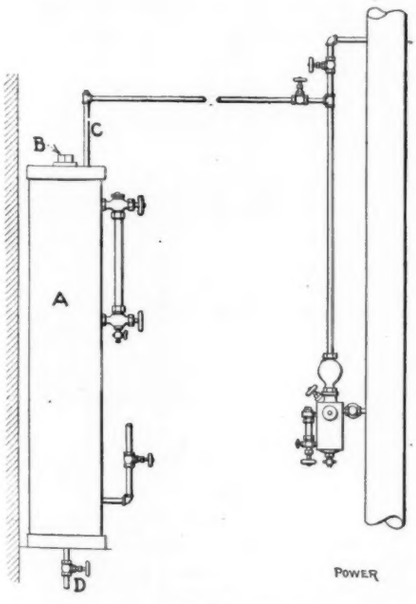
H. G. GIBSON.

Washington, D. C.

Filling Outfit for Lubricators

In a plant having many lubricators to be filled at comparatively short intervals the arrangement shown saves much time and trouble.

Alongside the pump or engine is located the oil reservoir *A*, made of a piece of 10-in. pipe of any length and fitted with caps at both ends. The top cap should be drilled and tapped to receive a plug *B* which can be removed for filling the tank. The oil outlet *C* to the lubricator should be ½- or ¼-in. pipe and connected to the top cap. The use of a glass gage is optional. A drain *D* should be provided for emptying the condensate.



OIL RESERVOIR FOR FILLING LUBRICATORS

If several lubricators are close together the filling outfit should be centrally located and the oil outlet piped to each lubricator. The manner of operation is evident from the illustration.

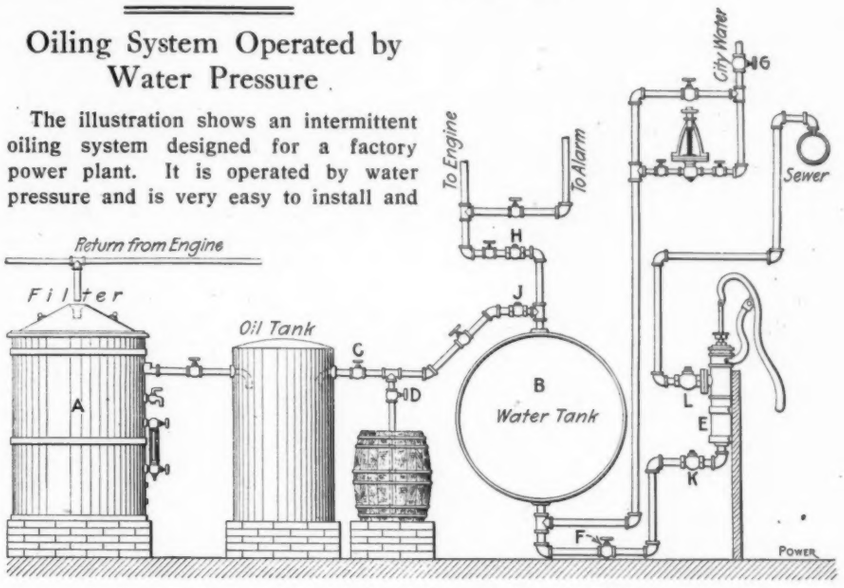
FRANK MCGOWAN.

Newark, N. J.

[The outfit might be further improved by connecting an oil line under gravity or forced feed to the filling hole of the reservoir.—EDITOR.]

Oiling System Operated by Water Pressure

The illustration shows an intermittent oiling system designed for a factory power plant. It is operated by water pressure and is very easy to install and use where it is possible to locate the filter *A* below the engines.



ARRANGEMENT OF OIL SYSTEM OPERATED BY WATER PRESSURE

With the storage tank *B* filled with water, and the water supply shut off, the valve *C* is opened to the receiving tank, or the valve *D* in the pipe shown between the two tanks, to a barrel of oil. The water is then pumped out of the storage tank by the hand pump *E* to the sewer and the oil flows into the storage tank as the water is pumped out. Next the valve *F* to the pump and the valve *C* to the receiving tank are closed and the valve *G* to the city water supply and pressure regulator is opened. The valve *H* is then opened to the engines and the oil, after passing through the bearings, flows by gravity to the filter, and from the filter to the receiving tank.

The engines are shut down at 5:30 or 6 p.m. and after all the oil has run from the filter to the receiving tank, the night engineer pumps the water out and the oil into the storage tank and the system is ready for operation next morning.

The receiving tank must be large enough to hold all the oil used during the day. Check valves *J*, *K* and *L* are fitted in the inlet and outlet pipes and also to the sewer pipe so that the oil cannot flow the wrong way or any be lost. If the water is turned on before the valve to the pump is closed, the water runs to the sewer and no harm can be done beyond the loss of the water. A gage-glass on the storage tank shows the amount of water and oil. Pressure and alarm gages and an alarm bell are located in the engine room. The pressure regulator is set to maintain a pressure of about 8 lb. more than is necessary to raise the oil to the engines, and the alarm gage is set so that the bell rings when the pressure falls to within 2 lb. of the pressure that will just raise the oil to the engines.

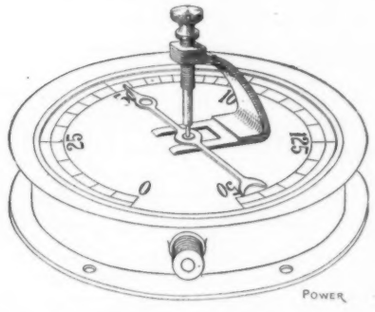
This system has been in constant service for over five years and has given no trouble.

JOHN HARPER.

Brooklyn, N. Y.

Removing Hands from Dial Gages

Removing the hands of most dial gages is often difficult. Unless great care is taken the tedious operation invariably results in bending the hand or defacing the dial and often bending the flanged front of the casing. To facilitate the work I use the device illustrated. It may be easily made of steel or brass. The illustration clearly shows its application to remove the hand.



GAGE HAND REMOVING DEVICE APPLIED

J. K. VOGLER.

Ellwood City, Penn.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Louisiana Eight-hour Law Criticized

In the Aug. 6 issue a news item covers the recent enactment of a law by the Louisiana legislature governing the hours of labor for engineers in that state. Your source of information was not trustworthy, because this law affects stationary firemen only, and but very few of them. This law seems to me to be the rankest kind of class legislation, and is in the opinion of high legal authority entirely unconstitutional.

The law as enacted applies to all businesses operating their power plants continuously night and day, except the petroleum industry, sawmills, cotton gins and presses, and sugar mills. Those familiar with the industries of Louisiana will perceive that practically only one business is affected by the law, i.e., public-service corporations. Anything not included in the above list of exceptions can probably without great inconvenience, shut down for at least one hour daily and so not violate the law.

If it is dangerous or undesirable, or even distasteful to the firemen to stand watches of 12 hours each in a plant that operates 24 hours daily, and if it requires an act of the legislature to rectify the "abuse," why does it still remain perfectly legal, to make one man stand a watch of 13 or 14 hours in a plant operating that long daily, but shut down the rest of the time? Furthermore, why not give the firemen employed in the excepted industries an opportunity to benefit by the law, also?

To understand my position in the matter one must realize that I have for three years been chief engineer for one of the corporations that "get it in the neck." I know of no fire room in Louisiana where white firemen are employed, except where oil is burned for fuel. I have paid my negro firemen, for hand firing coal, $2, $1.85, $1.75 and $1.50 daily for 12 hours, depending on the watch and the man. These wages and these hours may seem unreasonable to many in the North, but my plant has been considered a desirable one to work in because of the wages paid. Negro labor in the South is poor at best. The negroes are usually mentally deficient, irresponsible and they get "tired" and quit on a moment's notice.

Any man employing them has his hands full without any complications introduced by a paternal legislature. The law now says that I cannot work my negro firemen more than eight hours daily. Naturally, I figure that I cannot afford to pay them as much as formerly, particularly on the light watches, where about the only necessary qualification is the ability to remain awake, a natural difficulty with these firemen. But they will not "stand" for a wage reduction and instead, go to some plant where they can earn as much even if they have to work several hours more, for it is the high wages and not long hours they consider.

About the only effect of the law, as I see it, has been to shift a portion of the firemen's work and troubles onto the already overburdened back of the chief engineer, whose hours of labor, whose worry and responsibility are great enough without this new law to aggravate the situation.

JOSEPH POPE.

Meridian, Miss.

Substitute Piston Rings

L. M. Johnston's article under this heading in the Sept. 3 issue recalled an experience of mine. I was employed at a large hotel and, like most hotels, this one had a laundry in connection with it, which was busy every week day and until noon on Sundays. The laundry machinery was driven from the same engine that drove a 75-kw. generator, which was a 16x42-in. Corliss engine running at 100 r.p.m.

One Sunday after the engine had been acting badly, I removed the head and follower-plate and found one ring in seven pieces and about 3 in. gone from the other. We had no extra rings and as the laundry had to run on Monday, I substituted rings of 7/8-in. hydraulic packing.

When the engine was started on Monday morning plenty of oil was used in the cylinder and everything went along all right. I had expected to have a new set of rings made on Monday, and put in immediately, but the manager decided to put in a motor to run the laundry and have the engine overhauled. By the time the motor was installed, seven weeks had passed.

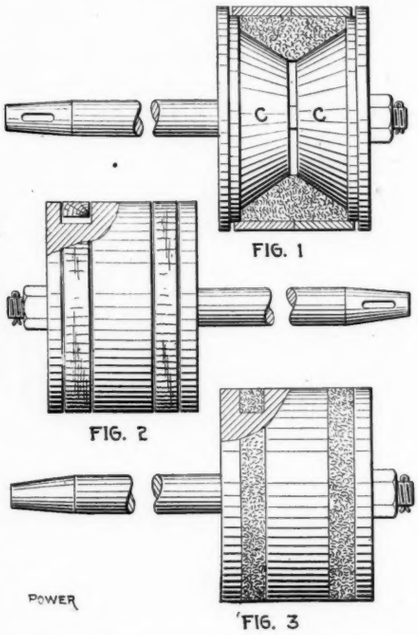
When the cylinder was opened, the packing was found in as good condition as when put in; it had worn smooth on the edge in contact with the cylinder.

ARTHUR V. LACY.

Julietta, Ind.

Mr. Johnston's article recalls my experience with substitute piston rings. Few engineers are acquainted with the old forms of pump packing, one of which is shown in Fig. 1. This was known as an improved hemp packing.

The piston had two broad packing rings made to the exact bore of the cylinder and had a follower-plate against them on each side which drew up the cones *AA* and set out the hemp filling, which expanded the rings. The piston nut could be tightened and a pin put through the rod to keep it from working off. Some of the heads had three or four screws to hold on these follower-plates. This kind of packing was much used formerly, and considered a big improvement over the old hemp packing which just filled the space between the flanges as it was easier to put in.



PISTON RINGS OF HYDRAULIC PACKING AND HICKORY WOOD

Fig. 2 shows another form of substitute packing, which has worked successfully for several weeks on small engines. The ring was made of a hoop of hard-wood which filled the ring groove without binding against the sides. I have seen such rings made from hickory wood that gave long and successful service.

I have corrected piston troubles by using a packing such as shown in Fig. 3. The rings were made of hemp rope

driven in the grooves, but they will not last long unless the piston head fits the cylinder bore closely. On steam pumps having broad steam-packing rings badly worn and leaking, I have filled in the grooves with hemp packing, and the pump consumed much less steam, as this hemp filling stopped the free passage of steam past the piston head. By putting sheet springs beneath badly worn snap packing rings they will be made steam-tight and run for several months without leakage.

C. R. McGAHEY.

Baltimore, Md.

Losses in Uncovered Steam Pipes

The recent letter under this heading, by J. L. Kezer, will inspire others to effect similar economies in conveying steam or hot water. That pipe covering and other insulation bring a much larger return for every dollar expended thereon than many other things in the plant is not sufficiently appreciated. Every opportunity for preventing loss of heat should be studied and taken advantage of if painstaking calculations prove that the expenditure is warranted. In Mr. Kezer's case the loss due to uncovered pipes amounts to $3160.90 per year.

Mr. Kezer will not be able to reduce the radiation loss to quite 90 per cent., as expected, for the steam pipe cannot be insulated with sufficient effectiveness. But the economic advantage in utilizing covering continuously is actually in excess of 200 per cent. per year as shown by the following calculations:

Assuming for simplicity, that all the steam pipe to be covered is of 2½ in. inside diameter, then the 3626.1 sq.ft. of radiating surface represents

$$3626.1 \times 1.329 = 4819 \textit{ lineal feet}$$

In testing nine kinds of commercial steam-pipe covering in 1901, Prof. D. S. Jacobus, of Stevens Institute, found that asbestos air-cell covering, saved 74.4 per cent. of the heat lost from bare pipe, which means in Mr. Kezer's case, a saving of steam worth

$$\$3160.90 \times 0.744 = \$2351.71 \textit{ per year}$$

Using asbestos-sponge felted covering the saving was found to be 84.9 per cent., which represents a saving of

$$\$3160.90 \times 0.849 = \$2683.60 \textit{ per year}$$

From these estimated savings remain to be deducted the first cost and depreciation charge. The air-cell covering, 1 in. thick, applied, may cost $500, and the asbestos-sponge felted covering $1000, not counting cost of covering fittings and flanges. Adding 10 per cent. for depreciation, the total cost of the air-cell covering during 10 years will be

$$500 + [10 \times (500 \times 0.10)] = \$1000$$

making in 10 years a net saving of

$$\$23,517 - 1000 = \$22,517$$

or a saving of 453.4 per cent. each year on the original investment of $500.

Using asbestos-sponge felted covering the cost in 10 years equals

$$\$1000 + [10 \times (1000 \times 0.10)] = 2000$$

making a net saving of

$$\$26,836 - \$2000 = \$24,836$$

or a saving of 248.4 per cent. each year on the original investment of $1000.

From the comparative results it does not follow that the use of high-priced covering is never justified because a highly efficient covering will further reduce the amount of condensation in a steam pipe feeding an engine and thus contribute to safety in operation.

A factor which tends to diminish the above difference in advantage between the two kinds of covering is found in the insulation required for valves, fittings and flanges. If, to be conservative, the radiating surface of these parts is assumed to equal 25 per cent. of the surface of the pipe, or 900 sq.ft., the cost of covering with either kind at 25c. per sq.ft. would be $225. This covering, owing to the use of cement, is credited with an efficiency of only 50 per cent. and its saving of heat in 10 years will amount to

$$900 \times 0.50 \times \frac{\$31,609}{3626.1 \textit{ sq.ft.}} = \$3924$$

Added to $23,517 the total saving in 10 years is $27,441. The total cost is

$$\$450 + 1000 = \$1450$$

leaving a gain of $25,991, which represents, with air-cell covering, a profit saving each year of 358 per cent. on the original investment of $725.

Now adding the same saving, $3924, to $26,836, the saving with asbestos-sponge felted-pipe covering, brings the total in 10 years up to $30,760. Deducting the total, $450 + 2000 = 2450 from this saving, leaves a net gain in 10 years of $28,310. The saving during each of the 10 years of life of the covering is therefore

$$\frac{\$2833 \times 100}{\$225 + 1000} = 231 \textit{ per cent.}$$

on the original investment of $1225.

These propositions are certainly attractive, but it is important to keep in mind, that the fewer the number of hours the covering is made use of, the smaller becomes the return from the investment, and *vice versa*.

Mr. Kezer's method of comparing the saving in radiated heat with equivalent tons of coal as delivered to the trestle is unusual. Since the efficiency of boilers will always be less than 100 per cent., more heat than that radiated from the pipes must be generated in the furnace. If the boiler efficiency is 70 per cent., the actual fuel consumption with uncovered pipes will be

$$\frac{2,719,575 \textit{ B.t.u.}}{10,817 \textit{ B.t.u.} \times 0.70} = 359 \textit{ lb. coal per hour.}$$

or 4.61 short tons per day, which at $1.65 = $7.61 per day, × 365 = $2777.65 per year, or 53.7 per cent. more than by Mr. Kezer's method.

Since by taking the cost of steam at the rate of 11.63c. per 1000 lb., the above sum is increased only from $2777.65 to $3160.90 or by 13.8 per cent. This steam cost evidently includes no fixed charges. In other words, the full cost of the steam is probably 15 per cent. greater than the net cost of 11.63c., and the saving capacity of the pipe covering is therefore greater than the calculations show.

CHARLES H. HERTER.

New York City.

Mr. Spencer's Belt Troubles

Mr. Spencer does not give the size of the pulleys in his letter on belt trouble in the July 9 issue, but I venture to say that the diameters of the driving and driven pulleys on which the rider gives trouble differ considerably while they are nearly the same on the other set of which he speaks.

Assume the engine-wheel diameter as 12 ft. and the driven-pulley diameter 4 ft. Neglecting creep and slipping the driven pulley makes $\frac{12}{4} = 3$ revolutions for one of the engine wheel. For each turn of this wheel the rim will travel $12 \times 3.1416 = 37.6$ ft. and the rim travel of the driven pulley will be the same.

The three-ply rider belt used by Mr. Spencer must be about ⅝ in. thick, which will cause it to act as a lagging on both pulleys, increasing their diameters

$$2 \times \tfrac{5}{8} = 1\tfrac{1}{4} \textit{ in.}$$

The rim travel with this increased diameter for one turn of the engine wheel will be

$$\frac{(12 \times 12) + 1.25}{12} \times 3.1416 = 38.0259 \textit{ ft.}$$

But the circumference of the driven pulley will be increased to

$$\frac{(4 \times 12) + 1.25}{12} \times 3.1416 = 12.893 \textit{ ft.}$$

and to make three turns for one of the driver it travels

$$12.893 \times 3 = 38.6793 \textit{ ft.}$$

during one revolution of the engine wheel. This amount is 0.6534 ft. more than the travel of the rider belt (without slipping) on the engine wheel, the rider therefore is drawn on by the driven pulley with a slip of 0.6534 + ft. per revolution of the engine wheel or one-third that quantity for every revolution of the driven pulley. Mr. Spencer will get better results if he leaves the rider off.

A friend informed me that he once riveted a rider to the under belt and after a run was surprised to find the rivets had pulled out. If Mr. Spencer will make a mark on the two belts which run without trouble, he will find after a run that the marks are not in the same relative position.

I recently saw a horizontal belt drive

(driving pulley 28 in., driven 84 in.) the 24-in. belt of which slips on the large pulley. The only reason I could see for this was that the small pulley had a very high crown, in fact, I have never seen another pulley with so high a crown. The belt has taken the shape of this crown so that it has troughed. The bottom of this trough does not touch the large pulley, but is carried around on a cushion of air and for this reason the slip comes on the large pulley, which is iron, the small one being paper.

L. A. FITTS...

Leominster, Mass.

The condition which Mr. Spencer mentions is due to two causes: first, that the original belt slips or creeps on the driven pulley, even with the rider on, and, second, that the old belt stretches more than the rider. In placing the rider belt, he probably drew it up very tight, it therefore stretches but little when loaded. There is little or no slip between the belts; hence the rider, in addition to increasing the friction on the pulleys, prevents the old belt from slipping as much as it did formerly. At first it would seem that less work is done with the rider on, because the effective pull of the original belt is lessened by the amount of tension in the new one. However, the rider increases the pressure along a 10-in. strip between the pulley and the big belt, with the result that the slip is not as great. Hence more work is done because, with the same belt speed, the driven wheel turns faster than before.

About the only thing that can be done with the drive as it stands, is to thoroughly clean the old belt with a mixture of equal parts of naphtha and benzine or some other grease-dissolving agent. Then, when the leather is clean, give it a thin coating of good belt dressing, applied hot and frequently in small amounts until the belt is soft and pliable. It will be as good as ever and will transmit as much power as possible for the given speed, width and tension. I had a similar experience, several years ago, but with a smaller belt. The difficulty was overcome in the manner described.

W. G. HAWLEY.

Clarks Summit, Penn.

I am not in favor of such a drive as Mr. Spencer is using when both pulleys are not of the same diameter. If he will figure out the speed of the shaft from the base pulleys, then add the thickness of the 30-in. belt to both pulley diameters he will perceive that the 10-in. belt tends to drive the shaft somewhat slower; therefore deriving its power from the driven instead of the driving pulley; hence the slack and tight sides of the belt in the position he states. The crowning of the pulleys will also make a difference in a drive like this one. If the driven pulley is the more crowned, it will aggravate the trouble; if the driving pulley is the more crowned, it may counteract the effect; therefore making it appear to be a satisfactory drive.

JAMES W. WILDE.

Pawtucket, R. I.

Centrifugal Pump Troubles

E. D. Gargnier's letter in the Sept. 3 issue under the above heading is good, but I would ask him what formula he would recommend to others who do not use a certain formula every day.

His formula

$$F = \frac{WRN^2}{35{,}196}$$

is all right if used frequently, but one who uses the formula for the centrifugal force only occasionally would spend more time hunting up this special one than by using the fundamental formula, and in this case the formula

$$F = Ma$$

will prove to be the shortest.

Since

$$M = \frac{W}{g}$$

and the acceleration a of the body toward the center is equal to $\frac{v^2}{R}$ it is evident that

$$Ma = \frac{Wv^2}{gR}$$

This is the formula which F. Webster used in the June 25 issue and which is easier remembered than the odd denominator in the special formula.

FRANZ SZABO.

Kewanee, Ill.

Phosphorus for Flue Gas Analysis

Allow me to call attention to Mr. Terman's articles on flue-gas analysis relative to the use of phosphorus as a reagent for determining oxygen. The phosphorus should be of the white stick form, and a pipette when filled with it looks very much like the pipette illustrated on page 330 of the Sept. 3 issue, the sticks of phosphorus occupying the positions shown for the glass tubes.

My reasons for preferring stick phosphorus to potassium pyrogallate are that the absorption of oxygen is quicker, that there is never the slightest doubt when a fresh supply of the reagent is needed, and that the pipette needs new reagent less often. One pound of the stick phosphorus is enough for several thousand analyses.

When the gas to be tested is admitted to the pipette, the sticks of phosphorus will begin to smoke. This will usually last only a minute or two and the reaction is over, and the gas should be run back into the measuring burette. The difference in volume is oxygen. A little time should be given to allow it to come back to the room temperature before measuring. No shaking of the pipette is required.

In handling the stick phosphorus precautions are necessary. If possible, handle only with pincers; if the fingers must be used, see that no particles adhere to them; give the phosphorus no chance to get under the fingernails, and keep the unused reagent under water. I have made several hundred analyses with both phosphorus and potassium pyrogallate, and I would not use the latter when the former was obtainable. White stick phosphorus must be used; the red will not do, nor will any other form than the sticks.

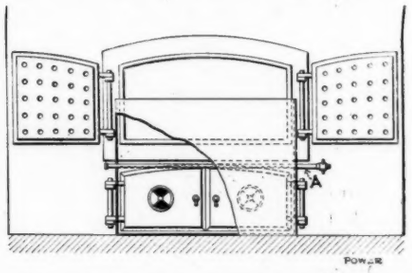
DONALD M. LIDDELL.

Elizabeth, N. J.

Device for Wetting Ashes

No doubt Mr. Chamberlain's method of wetting the ashes in the furnace as described in the Aug. 20 issue is quite effective for overcoming the ash dust in the boiler, but I think he sacrifices thereby considerable boiler efficiency.

The illustration shows an ash guard used in our plant to prevent dust from settling in the boiler room while cleaning fires. With this arrangement the



SHEET IRON BOX CONTAINING WATER PIPE USED IN WETTING ASHES

ashes are moistened at the time they are drawn from the furnace, thereby preventing the furnace from cooling as much as it would by Mr. Chamberlain's method.

The guard is a sheet-iron box shaped to the width of the furnace doorway. The front of the box is open so that as the ashes are pulled out they fall into the box and between it and the ashpit doors. A perforated water pipe A is run through the box so that the ashes may be wet as they are drawn out. The top of the box should be enough below the top of the furnace door to allow using the cleaning tools. If the box is made too deep, trouble will also be experienced in manipulating the tools when pulling ashes from the dead-plate. It should also set flush with the front of the boiler.

A. R. KUMMERER.

Hazelton, Penn.

Air Compressor Efficiencies

By E. M. Ivens*

Even when accompanied by a statement of terminal pressure and inlet temperature and pressure volumetric efficiency is nothing more than a guarantee of clearance volume.

Overall efficiency gives something definite and is a more satisfactory guide.

*Ingersoll-Rand Co., New Orleans, La.

Extending use of compressed air has brought into prominence such terms as "mechanical efficiency," "compression efficiency," "volumetric efficiency," and surprisingly few engineers actually know their real meaning.

"Volumetric efficiency" has become a high sounding catch word with compressor salesmen and manufacturers, and a high guarantee inserted in the specifications is regarded by many as a good and sufficient assurance of superior economy of the compressor. As a matter of fact, so many variables and limitations are embodied that a mere statement of the percentage of efficiency is not only misleading, but absolutely worthless when made without qualifications.

The simple formulas, derivations and discussions hereinafter given may be followed easily by keeping in mind the fundamental laws of expansion and compression of sensibly perfect gases.

Mechanical Efficiency

The energy in the steam admitted to the steam cylinder of an air compressor is expended:

1. To compress the air.
2. To heat the air during compression.

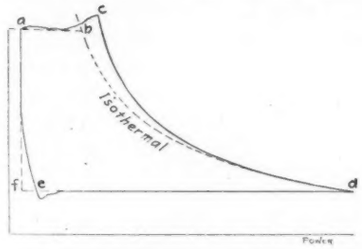


Fig. 1. Determining Compression Efficiency

3. To heat the jacket water.
4. To overcome the friction of the machine.

The horsepower required by 1 and 2 may be computed from the air-indicator diagram; 3 may be found by observing the temperatures of entering and leaving jacket water together with its weight and computing the heat units therefrom and the equivalent horsepower; and 4 is found by subtracting the sum of the first three from the indicated horsepower in the steam cylinder.

The mechanical efficiency of a steam-driven air compressor, then, is equal to the air horsepower plus the jacket horsepower divided by the indicated horsepower, or

$$E_m = \frac{A.hp. + jkt.hp.}{I.hp.} \qquad (1)$$

and the mechanical efficiency of a power-driven machine is expressed by

$$E_m = \frac{A.hp. + jkt.hp.}{B.hp.\ delivered\ to\ compressor\ shaft} \qquad (2)$$

This efficiency depends on the mechanical construction of the machine and the lubrication. It will be found to vary from 75 in poorly designed machines up to 92 per cent. in the best designs.

Compression Efficiency

Compression (or compressor) efficiency is the ratio of the theoretical horsepower required to compress an amount of air, to that actually required, or

$$E_c = \frac{Theoretical\ hp.}{A.hp.} \qquad (3)$$

The adiabatic and isothermal horsepowers are both theoretical, but since the term efficiency expresses how nearly perfect a machine or appliance is, the latter is proper to use (referring to no clearance base) in the formulas. This efficiency depends upon the design of the water jacket and cooling appliances, and it is principally to increase compression efficiency that multistage compression is employed.

To determine the compression efficiency, the isothermal curve is plotted on the air card, starting at the beginning of the stroke, and ending at the theoretical delivery line, or terminal pressure line. The area (Fig. 1) *abdfa* thus inclosed, divided by the area *acdea* of the actual card is the compression efficiency.

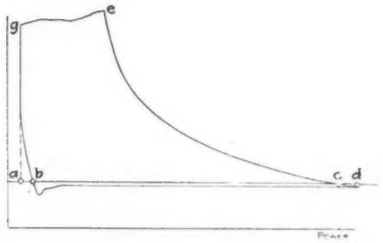


Fig. 2. Typical Compressor Diagram

Indicator cards that show a very high compression efficiency are open to suspicion as investigation will invariably show that either the suction valves leak, or air is escaping from the compression side of the moving piston to the suction side, due to a scored cylinder or leaky piston. Actual compression curves will follow the adiabatic quite closely as the water jacket has little effect other than to facilitate lubrication.

Volumetric Efficiency

Volumetric efficiency is the ratio of the actual cubic feet of free air compressed per unit of time to the cubic feet of piston displacement during that time, or

$$E_v = \frac{Actual\ cu.ft.\ free\ air\ per\ minute}{Cu.ft.\ piston\ displacement\ per\ minute} \qquad (4)$$

On the indicator diagram, Fig. 2, the observed volumetric efficiency is $ad \div bd$.

Volumetric efficiency depends, first, on the clearance volume in the air cylinder: With no clearance between cylinder heads and piston at the end of the stroke, and no lost space in and around the valves, the volumetric efficiency (referring to atmospheric air) would always be 100 per cent. The greater the clearance volume, the greater will be the volume of the cylinder occupied by the expanded clearance air. This is self-evident.

Volumetric Efficiency depends, second, on the terminal pressures: The higher the terminal pressure of air in any given cylinder, the greater will be the volume occupied by the expanded air of the clearance spaces. This means that as the terminal pressure is increased, the volumetric efficiency decreases. To show this graphically, three superimposed diagrams are reproduced in Fig. 3, taken from the same cylinder at the different pressures shown. The increase in volumetric efficiency as the terminal pressure decreases is plainly evident.

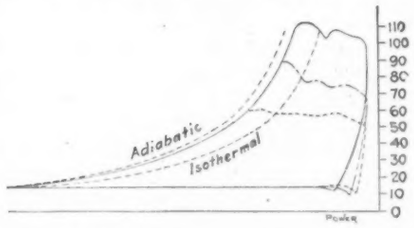


Fig. 3. Volumetric Efficiency Depends on Terminal Pressure

Volumetric efficiency depends, third, on the temperature and pressure of the intake air: Since, by definition, volumetric efficiency refers to free air, or air at 14.7 lb. pressure and 60 deg. F., every change of temperature and pressure of intake air has its effect directly on the volumetric efficiency. For instance, assume a room temperature of 60 deg. F. and atmospheric pressure at 14.7 lb., or conditions where actual free air is available and drawn without heating into the cylinder. In the room is a compressor whose capacity is 135 cu.ft. of this air per minute, whose piston displacement is 150 cu.ft. per min.; and the terminal pres-

sure 100 lb. The volumetric efficiency under these conditions is

$$E_v = \frac{135}{150} = 90 \text{ per cent.} \qquad (5)$$

Now, suppose that from some cause, the temperature of intake air were raised from 60 to 65 deg. F., the atmospheric and terminal pressures remaining as before. The compressor will still draw in 135 cu.ft. of air per minute, but owing to the higher temperature, a lesser weight or mass of air will be withdrawn from

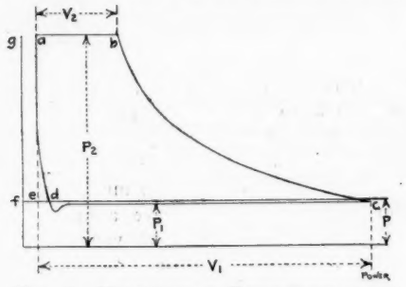


FIG. 4. VOLUMETRIC EFFICIENCY FROM INDICATOR DIAGRAM

the atmosphere. According to the law of Charles, 135 cu.ft. of air at 65 deg. F. and 14.7 lb. pressure is equivalent to 133.71 cu.ft. of air at 60 deg. F. and 14.7 lb. pressure. Under these conditions the expression for volumetric efficiency is

$$E_v = \frac{133.71}{150} = 89 \text{ per cent.} \qquad (6)$$

This shows that for every 5 deg. F. rise in intake air temperature, a loss of practically 1 per cent. in volumetric efficiency occurs.

Likewise, volumetric efficiency is affected by change of atmosphere or intake pressure—the temperature remaining constant. To show this, suppose the compressor removed to a higher altitude where intake air of 13.7 lb. pressure and 60 deg. F. is available. Now, 135 cu.ft. of this air is equivalent to 125.81 cu.ft. of free air and the third expression for volumetric efficiency is

$$E_v = \frac{125.81}{150} = 83.87 \text{ per cent.} \qquad (7)$$

Therefore, for every 0.163 lb. decrease of intake pressure, a loss of 1 per cent. in volumetric efficiency occurs.

FORMULAS AND MEASUREMENT

The most popular and convenient method of determining the volumetric efficiency of an air cylinder is from the indicator diagram. Referring to Fig. 4, the ratio $\frac{dc}{ce}$ might be called the observed volumetric efficiency. On the actual diagram, these distances are measured with some convenient scale, and the computations made and the results so obtained corrected for inlet temperatures and pressure. Volumetric efficiency found after these corrections are made is the true or real efficiency.

Expressions for real and observed volumetric efficiencies may be derived from the diagram as follows: Remembering that

$$PV^n = P_1V_1^n = \text{constant}$$

from Fig. 4 is obtained

$$\frac{df}{ag} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \qquad (8)$$

$$df = de + ag$$

Substituting in (8)

$$\frac{de + ag}{ag} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

whence

$$de = ag\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (9)$$

As ag is the clearance volume and may be determined by actual measurement by other approved methods, it is known, and designating it as V_c and substituting in (9), gives

$$de = V_c\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (10)$$

Now

$$\text{Observed } E_v = \frac{dc}{ce} = \frac{ce - de}{ce} \qquad (11)$$

where ce = piston displacement = P_d

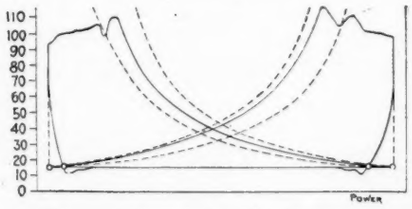


FIG. 5. EFFECT OF LEAKY VALVES

Substituting de as given by (10) in (11) gives

$$\text{Observed } E_v = \frac{P_d - V_c\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]}{P_d}$$

$$= 1 - \frac{V_c}{P_d}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (12)$$

Obviously the ratio $\frac{V_c}{P_d}$ is the percentage of the cylinder volume given up to clearance. Equation 12 shows that in scaling the diagram for volumetric efficiency the effect of clearance has been taken into consideration, and also terminal pressures, but not that of the initial temperature and pressure. To provide for this, (12) must be multiplied by $\frac{T_1}{T}$ where T is the absolute temperature of the air at the instant that compression begins, and T_1 is 60 deg. + 460.6 deg.

Also, the pressure at the beginning of compression is nearly always less than that of the atmosphere, due to frictional losses, valve-spring resistance, etc. To provide for this, equation (12) must be multiplied by $\frac{P}{P_1}$ where P is absolute pressure shown by the intake line on the diagram, and P_1 is 14.7 lb.

The expression for the true or real volumetric efficiency therefore is

$$\text{True } E_v = \frac{T_1P}{TP_1}\left\{1 - \frac{V_c}{P_d}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]\right\} \qquad (13)$$

Under some circumstances, this method of volumetric efficiency determination is to be avoided, for results then obtained will be misleading and consequently worse than worthless. Leaky suction valves or stuffing-boxes, and a cylinder scored at or near the end of the stroke will produce an almost perpendicular re-expansion curve.

These defects may be detected on the diagram by plotting the theoretical curves and comparing with the actual. Fig. 5 shows the typical case of leaky valves on one end and their effect on the volumetric efficiency and comparison curve.

A better way to determine the volumetric efficiency of a compressor under all conditions of cylinder, is to actually measure the air delivered and divide by the piston displacement. The air may be measured by a standard orifice or by a system of inclosed tanks. The former method has been frequently described. The latter is as follows:

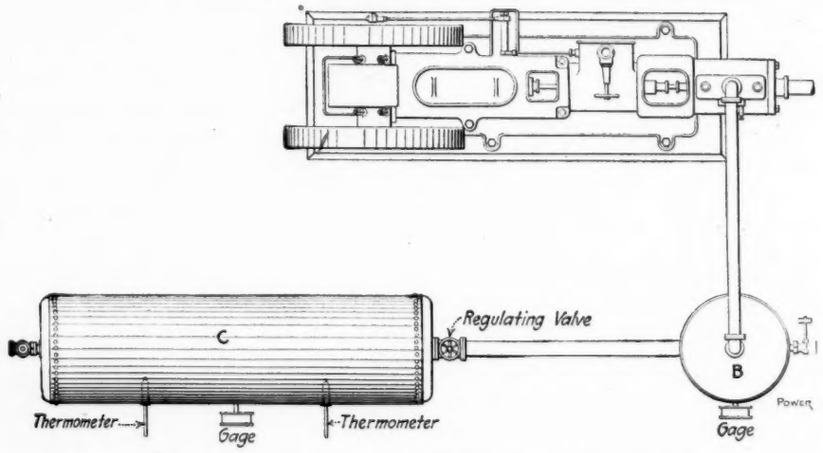


FIG. 6. MEASURING AIR DELIVERED TO COMPRESSOR

Connect the air compressor to two inclosed tanks, B and C, as in Fig. 6, with a regulating valve between B and C, air gages and thermometers as shown.

Through the regulating valve, the air pressure in B may be maintained at the desired pressure for which the volumetric efficiency is to be determined. The tank C then may be pumped up from pressure P_1 to P_2, both lower than the pressure in B. It is always advisable to begin the test at an initial pressure higher than atmospheric. Shut off tank C and start the compressor. Watch the gage on C until the needle reaches a suitable point, say 15 lb., and from then on, until pressure P_2 is reached, count the revolutions of the compressor and observe temperatures and the time of the run. A number of runs should be made and the mean of the results found and substituted in the formula following:

Let

P = Atmospheric pressure = 14.7 lb.;

P_1 = Initial absolute pressure in tank C;

P_2 = Final absolute pressure in tank C;

T = Absolute room temperature in degrees F.;

T_1 = Absolute initial temperature of air in tank C;

T_2 = Absolute final temperature of air in tank C;

V = Volume in cubic feet of tank C;

V_1 = Free air equivalent of air in tank at beginning of test;

V_2 = Free air equivalent of air in tank at end of test;

V_3 = Actual amount of air pumped into tank;

$R_1 = \frac{P_1}{P}$ atmospheres beginning of test;

$R_2 = \frac{P_2}{P}$ atmospheres end of test.

Disregarding temperature,

$$V_3 = V_2 - V_1 = \textit{Volume of air compressed}$$

$$V_1 = V\frac{P_1}{P} \text{ and } V_2 = V\frac{P_2}{P}$$

$$V_3 = V\left(\frac{P_2}{P} - \frac{P_1}{P}\right)$$

$$V_3 = V\ (R_2 - R_1) \qquad (14)$$

The theoretical quantity of air pumped is equal to the piston displacement of the compressor, or $l \times a \times n$, where l = length of stroke in feet, a = area of the piston in square feet, and n = number of strokes. The observed volumetric efficiency is then represented by the formula

$$\textit{Observed } E_v = \frac{V\ (R_2 - R_1)}{lan} \qquad (15)$$

Correcting for temperatures, (14) becomes

$$V_3 = VT\left(\frac{R_2}{T_2} - \frac{R_1}{T_1}\right) \qquad (16)$$

If the barometer pressure is other than 29.92 in. of mercury, the formula should be corrected and made to read as follows:

$$V_3 = \frac{VT}{B}\left(\frac{29.92\ R_2}{T_2} - \frac{29.92\ R_1}{T_1}\right) \qquad (17)$$

The expression for real volumetric efficiency then becomes

$$\textit{Real } E_v = \frac{\frac{29.92}{B}\left(\frac{R_2}{T_2} - \frac{R_1}{T_1}\right)}{lan} \qquad (18)$$

The definition and formula derivation have shown conclusively that unless a guarantee of volumetric efficiency is accompanied by a statement of the terminal pressure and inlet temperature and pressure, it is meaningless. If these conditions are stated, the guarantee is nothing more nor less than a guarantee of clearance volume. So, really, what excuse is there for the existence of the term "volumetric efficiency?"

Overall Efficiency

This efficiency is the most important of all to the user, for it refers directly to the cost of operation. It means the cost in fuel, upkeep, supplies, interest and depreciation of air delivered to the receiver or pipe line, and is consequently a combined statement of mechanical compression and volumetric efficiencies as well as reliability.

The following is a general expression of overall efficiency referred to the isothermal no clearance base.

$$E_o = \frac{\left(\textit{Boiler hp. per 100 cu.ft. free air per min. actually delivered}\right)}{\left(\textit{Isothermal hp. per 100 cu.ft. free air per minute}\right)}$$

This expression does not take into consideration all the factors that affect overall efficiency, but even as it is, it is something definite and a much more satisfactory guide than any yet given. Elaborate tests over long periods of time are necessary to determine the true overall efficiency of any machine and no accurate formula can be derived that will take into consideration even one of the variables, such as upkeep or depreciation. The most reliable information on the subject is a series of tests made by Richard L. Webb on a number of air compressors in the Canadian mining district, published in "Compressed Air Plant" by Prof. Robert Peele.

Economy Essentials

The foregoing discussion shows that the economy of an air-compressor unit, depends:

(1) On the mechanical construction, that is, the size and proportion of bearings, and wearing surfaces; lubricating system and general design of parts.

(2) On the length and volume of ports in the air cylinder. Long and tortuous ports and air passages increase the losses by heating the incoming air before compression begins.

(3) On the cooling devices and water jackets and temperature of cooling water.

(4) On the surrounding conditions, that is, altitude at which the machine is being operated and atmospheric temperature.

(5) On the clearance spaces.

(6) On economy of power end.

Weight of Mercury

By C. N. Cross

Mercury is very frequently used for measuring pressures varying from 0.1 to 4 or 5 lb. per sq.in. and occasionally as high as 150 to 200 lb. in steam-gage calibration. It is used almost exclusively to determine the vacuum in condensers, so that it is important to know the weight of mercury in pounds per square inch, and to know how the weight or density varies with change of temperature. The errors introduced by neglecting to correct for changes of density are much smaller for steam pressures above atmospheric, than when measuring high vacuum.

For accurate pressure measurement mercury should be pure, as impurities alter its density. Impure mercury shaken with air yields a black powder, caused by the oxidization of the impurities. Another indication is the "tail" which is seen when a drop of the metal is run over an inclined surface. Pure mercury has a silver-white color and globules retain a perfectly spherical shape. The surest way to free it from impurities is to distill it. Its boiling point is about 675 deg. F. It may be more easily purified by treating it with a 5 or 10 per cent. solution of nitric acid. The mercury is run through a very small tube into the vessel containing the nitric acid, and the treatment may have to be repeated several times.

The weights of mercury in the accompanying table, which have been transposed from metric to English units, were determined by Thiesen and Sched.

WEIGHTS OF MERCURY AT DIFFERENT TEMPERATURES

Temperature Deg.	Weights of Mercury lb. per cu.in.	Correction Factor to Reduce to 32 deg.F.
32.0	0.49116	1.0000
39.2	0.49080	1.0008
50.0	0.49028	1.0018
59.0	0.48980	1.0028
62.6	0.48964	1.0032
68.0	0.48932	1.0037
77.0	0.48895	1.0045
86.0	0.48850	1.0053
95.0	0.48803	1.0064

As the third column shows, the density of mercury decreases about 0.64 per cent. when the temperature is increased from 32 to 95 deg. F. From the second column the following values have been computed. At 62.6 deg. F., a column of mercury 1 in. high produces a pressure of 0.48964 lb. per sq.in., 7.834 oz. per sq.in., 13.570 in. of water at 62.6 deg. F., 70.502 lb. per sq.ft., 1.1308 ft. of water at 62.6 deg. F., 927.95 ft. of air. The last value is for dry air at a pressure of 29.921 in. of mercury and 62.6 deg. F.

Dundee Meeting of British Association

By John B. C. Kershaw

A review of the papers bearing upon power generation and relating to heat flow from gases into cylinder walls of gas engines, coal mine explosions, new bomb calorimeter, gas turbines and heat loss from bare pipes.

On Sept. 4 to 11, the eighty-second annual meeting of the British Association for the Advancement of Science was held at Dundee, Scotland. Five papers bearing upon power generation were contributed to the engineering section of the association, and these are abstracted in the following:

Efficiency vs. Nuisance

The presidential address to the members of the engineering section, by Prof. A. Barr, centered on the duty of the engineer to pay more attention to artistic considerations in the pursuit of his calling, and to practice the profession with more regard for the material interests and aesthetic susceptibilities of the general public. According to Professor Barr, when one examines into the immediate causes of the injuries and inconveniences that result from the activities of engineers, one finds that they are due in almost all cases to failures rather than to successes. The more completely the engineer achieves the primary end of his work the less is the damage or injury that can be laid to his charge. Following up this idea, Professor Barr discussed the subject of power generation from solid fuel in some detail, and gave it as his opinion that the present wasteful methods would soon give place to others that would be both more scientific and less harmful to the community. Following are some extracts from his address:

Progress towards economy is leading to a reduction, and possibly to the entire elimination of all the nuisances associated with the older methods of power and heat production. The great improvements that have recently been made in producer plants and gas engines have rendered out of date, as regards economy, at least the smaller sizes of steam plants which are so fruitful a source of injury and inconvenience to the community; and we now have engines of the Diesel, and the so-called semi-Diesel types, that can utilise natural oils, and oils obtained in the distillation or partial distillation of coal, not only with an efficiency hitherto unattained in heat engines, but "without injury or damage to any one single person"—except possibly the maker of inferior plants.

Present indications point to the coming of a time in the near future, when the power and heat required for industrial and domestic purposes will be distributed electrically in a perfectly inoffensive manner from large central stations; and even at these stations there will be no pollution of the atmosphere that could give the most sensitive of critics any just grounds of complaint against the intrusion of science into our lives.

It is just a hundred years since passengers were first carried on the Clyde in a mechanically propelled ship, and today —when they are not too completely obscured by smoke—we can see the successors of the 'Comet,' plying on that river with power plants of greatly superior overall efficiency but showing little advance in regard to the combustion of the fuel. Had the emission of smoke from river craft been prohibited years ago, there is little doubt that engineers would have let few days pass without arriving at some solution of the problem of inoffensive power production on steamboats, and the demand for economy would have looked after itself. How much better it would be were engineers to take the wider view of their duties and responsibilities and realise that they were acting contrary to the true spirit of their profession, when they produce appliances that pollute the atmosphere for miles around, to the hurt and inconvenience of those whose use they are intended to serve. But this year a ship has left the Clyde that we hope may be the forerunner of a new race, which will attain a higher efficiency than any of the direct descendants of the 'Comet.'

Gaseous Explosions

The fifth report of the committee appointed by the British association to investigate "Gaseous Explosions," with special reference to the temperatures attained in internal-combustion engines, was presented by Dugald Clerk. In the present report, the committee has dealt chiefly with the question of the rate of heat flow from the gases, or working fluid, into the cylinder walls. The report states that it is now fully recognized that a great part of the difficulties experienced in the construction and working of internal-combustion engines is due to heat flow, and that the subject has been brought into greater prominence in recent years by the construction of larger engines and cylinders in which these difficulties have only partially been overcome.

The important practical question is the mean rate at which heat goes into each part of the surface, and the resulting mean distribution of temperature. The chief problem in designing large gas engines is to control the mean temperature distribution by water jacketing or otherwise in such a way, that the metal does not get overstrained by unequal expansion, nor reach a temperature sufficient to ignite the gas. The temperature gradient necessary to sustain the flow of heat from the inside of a combustion chamber to the external water is not likely to exceed 120 deg.F. per inch. At places where the metal is not thick, and effective external circulation of water is possible, cooling does not present great difficulty; but at places which are not near to the cooling water, so that the heat has to travel a long way, the temperature must be high to give the necessary gradient. Thus the central portion of the head of an ordinary flat-faced piston, if not water cooled gets very hot, reaching a temperature of perhaps 1100 deg.F. in a four-cycle engine of 24 in. bore. The piston expands considerably in consequence, the expansion being greater at the center than at the edge, which is accordingly put into tension. In larger cylinders the stresses in the piston set up by unequal heating, and the danger of pre-ignition arising from the hot metal, necessitate the cooling of this part by the circulation of oil or water. Even then the greater thickness of metal in certain parts of the combustion chamber, and the difficulty of keeping the water flowing properly in every corner, may cause high local temperatures.

The heat carried away by the cooling water and by radiation, is the total given to every part of the walls, and its measurement gives no information on the important question of the manner in which the flow is distributed over the walls. It is certain, however, that the greater part of the heat flow in a cycle occurs in a comparatively short time just after the moment of ignition, and passes therefore into the surface of the combustion chamber and valves and into the face of the piston. But little goes into the barrel of the cylinder which is not uncovered, until the density and temperature of the gases have fallen. That this must be so is obvious, but the magnitude of the effect is perhaps not generally recognised. Dugald Clerk found in his experiments on the compression and expansion of flame that the average heat flow per square foot per second in the first three-tenths of the stroke, is three times that of the average over the whole stroke for equal temperature differences, and he calculates that the actual rate of heat flow in the first three-tenths, is six times that of the whole stroke in ordinary gas engines, working at full load. This estimate, however, does not include loss due to radiation before maximum temperature. In the actual firing and expansion stroke of a gas engine the difference must be even more when radiation and other losses incurred before maximum temperature are included, and it is probable that in discussing the problem of cooling the metal, it is a sufficiently good approximation to neglect the heat flow into the outer half of the barrel altogether. Professor Hopkinson infc ns the committee that he has worked a gas-engine cylinder of over 30 in. diameter, in which there was no water circulation round the barrel at all. The whole of the heat passing into the barrel was in this case removed either by radiation or by conduction into the piston, nor was the cooling which was applied to the piston much more than that found necessary on other parts of the walls of the combustion chamber. In small engines with uncooled pistons, the water jacket around the barrel is necessary to keep the piston cool.

In the scientific analysis of gas-engine phenomena, the facts stated in the last paragraph are important, because they show that the heat flow is not much different from that which would occur in a closed vessel of invariable volume having the form and size of the combustion chamber, the mixture fired having of course the same composition, density, movements, etc., as in the engine. Some

allowance must be made for the fall of temperature and density which occurs in the initial stages of the expansion in the engine, but this will be of the nature of a correction, and will not affect the value of the general conclusions as to the effect of the various factors in heat flow which may be drawn from closed vessel experiments.

The report then discusses in detail the various factors that affect heat flow, these being enumerated as follows: The state of cleanliness and brightness of the internal walls, radiation from the gas, effect of cylinder dimensions, effect of density and effect of turbulence. The last-named effect is one discovered by Dugald Clerk, and is due to the eddying or turbulent state of the gases within the cylinder, following upon the suction stroke of the engine. In consequence of this turbulent state of the gases, the explosion occurs more rapidly than in a quiescent gas mixture, and the effective heat conductivity of the gas is increased. Dugald Clerk has worked out a practical method of measuring and recording this increased conductivity, and the report contains two diagrams illustrating his method and its results.

Prevention of Coal Mine Explosions

Professor Dixon, of Manchester, followed Dugald Clerk with an account of the work carried out at the government's experimental station, for investigating the causes and prevention of coal-mine explosions, located at Eskmeal, near Scawfell in Cumberland. This plant includes an explosion gallery 900 ft. long and 7.6 in. in diameter, with similar galleries of smaller dimensions, and apparatus for charging these galleries with any desired gas mixture and for observing the force of the explosions produced by firing the mixture. So far as the experiments have gone, they prove that the best way to prevent dust explosions in mines, is to use some incombustible dust, with which the combustible coal dust must be covered in every gallery and other place where it collects.

Incidentally the experiments have proved the correctness of Dugald Clerk's observations on the effects of turbulence upon the speed and force of the explosive wave, a small fan being introduced into the gallery to create the desired turbulent effect. The most destructive explosions were obtained when this fan was run at a high speed to churn up the dust and gas mixture.

New Bomb Calorimeter

A paper, of which the following is an abstract, was read by Robert S. Whipple upon a new form of bomb calorimeter for testing the calorific value of coal and other forms of fuel:

The bomb form of calorimeter is now frequently employed to determine the calorific value of coal, because the combustion of the coal is more complete than in those calorimeters in which the oxygen is admitted at atmospheric pressure or slightly above.

The instrument designed by Prof. C. Fery, of the Ecole de Physique et de Chimie, Paris, is of the bomb type, being in general design somewhat similar to the Mahler instrument, but greatly simplified in its details. The bomb consists of a light iron vessel weighing about 2.2 lb., and having a capacity of about 15 cu.in. supported in the center of a brass vessel by two disks of constantan (an alloy of copper and nickel). As the mass of metal in the bomb is small and no water jacket is used, the rise in temperature of the bomb due to the combustion of the coal is large, averaging about 36 deg.F. when 0.5 gm. of coal is burnt. It is evident that with such a large temperature rise the measurements need not be made with the same degree of accuracy as in the case of the usual type of bomb calorimeter, in which the temperature rise amounts to only 4 or 5 deg. F. for the same quantity of coal.

The temperature rise is measured by the thermo-electric force generated by the constantan disks and the iron bomb, the surrounding envelop acting as the cold junction of the thermocouples. These couples give an electromotive force of 22.2 micro-volts per degree F., giving on a pointer galvanometer a deflection of 2.4 in. for a rise of 36 deg.F. It is thus possible to estimate the temperature rise of the bomb to $\frac{1}{100}$ of the total rise in temperature.

The loss of heat from the bomb is due to three causes: (1) conduction along the constantan disks; (2) convection currents in the air space surrounding the bomb; and (3) radiation from the walls of the bomb.

The effect of the loss by conduction is only to lower the maximum temperature obtainable as compared with what it would have been if the supporting disks were nonconductors. The losses due to convection and radiation are negligible as compared to those due to conduction.

The coal under examination is placed on a small tray in the bomb, the latter being filled with oxygen at about 200 lb. pressure. The coal is fired electrically, the maximum temperature rise on the galvanometer taking place about 1¼ min. after ignition.

Experiments show that the gas pressure can be varied from 150 to 250 lb., and the weight of coal burnt from 0.2 to 0.7 gm., without any difference in the results obtained. The instrument is standardized by burning either sugar, carbon, or standardized coal briquets, the calorific values of which are known.

Gas Turbines

On Sept. 9, an interesting discussion upon "Gas Turbines" was opened by a paper from Dugald Clerk. The author stated that engineers engaged in the development of the internal-combustion engine, have long recognized that great advantages would come from substituting rotary for reciprocating movement, and accordingly many attempts have been made to produce a commercial gas turbine. So far, no attempt has succeeded; the practical difficulties have proved too serious. Much useful knowledge, however, has been obtained by able experimenters, and we are now in a position to consider the difficulties afresh, with some experimental data at our disposal.

Details of the turbines and work of Armengaud, of Karovodine and of Holzwarth were given.

The Armengaud turbine was of the constant-pressure type, and the largest experimented with gave 300 b.hp. with a fuel consumption of 3.9 lb. of petrol per b.hp.-hr. This was nearly eight times the petrol consumption of the ordinary internal-combustion type of engine, and was much too high for this turbine to be considered as a commercial engine.

The Karovodine turbine was of the explosion type without compression, and of exceedingly ingenious construction. It contained four explosion chambers, having four jets actuating a single turbine wheel of the De Laval type, about 6 in. in diameter, and running at 10,000 r.p.m. This turbine gave 1.6 b.hp. with a fuel consumption of 6.5 lb. petrol per b.hp.-hr., and thus consumed more than double the petrol used by the larger constant-pressure gas turbine of Armengaud.

The Holzwarth gas turbine was also operated on the explosion principle, but the construction and action were very different from that of Karovodine. The turbine in general arrangement outwardly resembled the Curtis steam turbine, in that the turbine wheel rotated in a horizontal plane, the spindle or shaft was vertical, and a dynamo was mounted on this spindle above the turbine. In the Holzwarth turbine ten combustion chambers were provided, each of a pear or bag shape. They were arranged in a circle around the wheel, and were cast so as to form the base of the machine. The wheel was of the Curtis type, with two rows of moving and one of stationary blades.

The inventor has built a gas turbine of a rated capacity of 1000 hp. of this type, the wheel of which rotates at 3000 r.p.m. Many practical difficulties have been met with in the selection of a gas suitable for the operation of this turbine, and ultimately the turbine was worked with a producer gas made from coke. The maximum power obtained from the turbine in the early experiments was 160 b.hp., with all ten explosion chambers in operation, and an explosion pressure in the chambers of 45 lb. above atmospheric pressure was obtained. It was found, however, that higher explosion pressures and a greater horsepower per chamber could be obtained with fewer chambers in operation. With five chambers, a total of 145 hp. was obtained, at 29 hp. per chamber; and with four chambers, the total was 121.6 b.hp. at 30.4 hp. per chamber. With 10 chambers in operation the power fell to about 15 hp. per chamber. From these experiments, Mr. Holzwarth concluded that the successive explosions interfered with each other and prevented proper charging and action.

The latest results with this turbine (communicated by Mr. Holzwarth to Dugald Clerk by telegraph, specially for

the meeting) have shown a total power production of 450 hp. with this turbine, and an efficiency of 23 per cent., but it is not clear upon what basis this efficiency is calculated.

Mr. Clerk in closing his paper, summarized the present position with regard to the gas turbine, as follows:

> The existing internal-combustion engines are quite satisfactory for small and moderate power units; but the weight increases so rapidly with increase of cylinder diameter that large units such as 20,000 hp. per shaft, easily attained by the steam turbine, have proved quite impossible for the reciprocating gas engine. To apply internal combustion for the purpose of such large units, it appears necessary to dispense with the cylinder piston and crank. I fear that this cannot be done on the lines of either constant-pressure or explosion turbines here shortly discussed. The results obtained only appear to show that progress can hardly be expected on the lines of flame impinging on turbine blades, either in impulse or reaction turbines. Several engineers have suggested the use of explosion to give water velocity, which velocity actuates a water turbine of some form. Mr. Humphrey, in his paper to the Institution of Mechanical Engineers, figures such a combined gas and water turbine; and although in the form suggested by him the conditions required too cumbrous a machine, yet it does seem that the more hopeful line is to use explosion and expansion to give water velocity, and so avoid all heat difficulties in the turbine part of the apparatus. True, such an arrangement still necessitates a reciprocating mass of water, but it will probably be found that great gain in weight can be obtained by the suppression of the piston, connecting rod, engine frame and crank. This line seems to me to be a much more hopeful one than any scheme involving the direct contact of flame with turbine blades.

Sir William White, who opened the discussion, emphasized the advantages of the rotary principle for engines designed for ship propulsion, and gave as examples the comparatively small space (relative to the ship's tonnage) occupied by the enormously powerful engines of torpedo destroyers, and of the "Mauretania" and "Lusitania," and also the freedom from vibration of these ships, as compared with ships of similar size and power, driven by reciprocating engines. Sir Charles Parsons was experimenting on the subject of gas turbines, but the difficulties enumerated by the author of the paper had so far blocked the way to all further progress toward the design of a successful and economical gas turbine.

Prof. Sylvanus Thompson referred to the upper and lower temperature limits (1800 deg. C. and 400 deg. C.) given for gas-turbine work in the course of the paper, and thought that these could be improved upon. The use of carborundum for lining the explosion chambers of gas turbines did not seem to him a practicable means of protecting the metal from the hot gases, for the carborundum would be certain to crack and become disintegrated, with disastrous results to the turbine blades. Mr. Ferranti was stated to be carrying on experiments with a new type of gas turbine, and Professor Thompson would have liked some information as to the progress he had made in solving the difficult problem.

Sir Alfred Ewing thought that the difficulty as regards temperature would be overcome by the discovery of a metal (or material) which would prove durable at temperatures of 1800 deg. C. or over, and could be used for cylinders and explosion chambers of gas engines and gas turbines.

Dr. Robert Mond gave details of the experiments made by Humphreys some years ago, with the combined gas and water turbine, and stated that these experiments failed since this turbine required another engine to operate it.

F. Samuelson defended the steam engine and steam turbine as prime movers against the attack of gas-engine and gas-turbine makers, and was convinced that much higher efficiencies than had yet been obtained could be reached with steam turbines, if steam pressures up to 400 lb., and steam temperatures of 750 deg. F. were used for their operation. This would increase the practical efficiency of the turbine by 20 per cent.

Dugald Clerk in his reply to the discussion explained that such high temperatures as the last speaker advocated for steam turbines were impracticable, since there is some alteration in the physical structure of the iron which caused its swe ling or "growth" when exposed repeatedly to temperatures of 700 deg. F. Mr. Parsons had convinced himself and the scientific public that the only practicable method for using higher temperatures than are at present customary in steam-turbine practice, was to utilize the first pressure and temperature drop, in the impulse type of turbine.

As regarded the position of the gas engine and gas turbine in face of the growing competition of the Diesel type of motor, he pointed out that the check to any great expansion of the use of oil engines, was the comparatively limited supply of oil fuel.

The total world's output of coal was now in the neighborhood of 1100 million tons per annum; while the total oil production was only about 45 million tons. From this latter total must be deducted the oil required for illuminating and lubricating purposes, also that for motor vehicles and aëroplanes. The balance represented the oil for power and heating purposes, and as compared with the 1100 million tons of coal, it represented a very small proportion of the world's fuel supply. The makers of steam and gas engines need not be apprehensive therefore as to the growing competition of the oil-driven type of engine, and so long as cheap coal supplies are available the future of their industry was safe.

Heat Loss from Bare Pipes

An interesting paper upon the "Transmission of Heat from Steam-heated Pipes through Conducting and Nonconducting Materials" was communicated by C. H. Lander and J. E. Petavel. The results of this investigation were important in their bearing upon the choice and application of heat-insulating coverings for steam pipes and boilers.

The rate of flow of heat from a hot metal cylinder depends on the temperature of the cylinder, its diameter, and the pressure of the air surrounding it. A number of measurements have been made, using cylinders ranging in diameter from 0.02 in. to 12 in., the smallest ones being heated electrically up to a temperature of 2000 deg. F. while for the larger ones steam heating was adopted.

Many of the radiating cylinders were surrounded by an air-tight inclosure, and the pressure of the air within this was varied from atmospheric pressure up to 3000 lb. per sq.in. The part of the work bearing on the loss from steam pipes of ordinary sizes may be summed up as follows:

Measuring the heat loss per hour in British thermal units per square foot per degree Fahrenheit of temperature difference, with saturated steam at atmospheric pressure and 100 lb. gage, the external temperature being 60 deg. F., the results are shown in the accompanying table.

Under these conditions the heat loss is mainly due to convection. Taking as an example a pipe of 1-in. external diameter with emissivity of 3 B.t.u. for steam at atmospheric pressure, the loss by radiation will be 0.44, or say, 15 per cent.

The loss by conduction in an inclosure 8 in. greater in diameter than the pipe, figures out at 0.13, or about 4 per cent. and would be rather less in an inclosure of larger diameter. Thus convection accounts for 81 per cent. of the loss from a 1-in. pipe containing steam at atmospheric pressure, and a similar reasoning would lead to the conclusion that it is about 80 per cent. for steam at 100 lb. pressure. For very small diameters below 1 in. conduction becomes relatively more important, and may even amount to one-half of the total.

The figures refer to an oxidized surface, such as that of the ordinary unpainted steam piping. Careful polishing would diminish the amount lost by radiation to one-quarter, while machining the pipe to an ordinary finish, would halve the radiation.

In the first case, the loss from the pipe containing steam at atmospheric pressure, would decrease to 2.67 and the convection would be 91 per cent. of the total, in the second the total loss would decrease to 2.78 and the convection would be 87 per cent. of the total. As an illustration the 2-in. radiator used in the above work, lost 2.55 B.t.u. per deg.

when coated with lampblack varnish, this being reduced to 2.32 B.t.u. after cleaning.

The object attained by surrounding a pipe with insulating material is to check convection currents and thus eliminate or diminish the loss due to this cause, the insulating material also reduces radiation.

HEAT LOSS FROM METAL SURFACES SUCH AS BARE STEAM PIPES OF VARIOUS DIAMETERS

External dia. of radiator in.	Heat loss per hr. in B.t.u. per sq.ft. per deg. F.		Conduction only, taking K = 144*	
	0 lb. gage	100 lb. gage		
0.024	9.20	10.5	3.4	Measured in an enclosure
0.043	6.21	6.90	2.2	0.81 in. dia.
0.20	3.94	6.0	0.39	Assuming layer of air to be 4 in. thick round radiator
1.00	3.00	4.0	0.13	
2.00	2.55	2.8	0.089	
3.00	2.30	2.4	0.074	
5.00	2.05	2.1	0.060	
12.00	1.97		0.047	
Flat layer 4 in. thick.			0.036	

*The constant K is given in B.t.u. per hr. per sq. ft. of surface, per deg. F. Temperature difference, for a layer 1 in. thick.

All solid materials used as insulators are better conductors than air, and using an insulating material, therefore, really increases the conduction loss. For instance, with a wire of small diameter, in air at low pressure, conduction is more important than convection, and an insulating covering will actually increase the thermal loss.

Even with a 1-in. pipe, a material such as slag-wool or asbestos if applied in a solid compressed block, would increase the total loss. The insulating effect obtained depends on so arranging the substance that the least quantity of material will subdivide the space around the pipe into separate air cells as perfectly as possible. Thus the efficiency of slag-wool is at a maximum when the solid material occupies about one-fourteenth of the space. Under these conditions the loss by conductivity is increased from 0.13 to 0.40, but assuming radiation to be eliminated, the loss by convection is reduced from 3.2 to 0.14 and the total loss is only about one-seventh that of the bare pipe.

When the material is either more closely or more loosely packed, the total loss increases; in the former case because of the increased conductivity of the solid material, in the latter case because of the greater loss by convection.

In certain substances, such as cork or wood, the natural arrangement of the material is cellular, and as such, satisfactory from the point of view of heat insulation.

Finally, with regard to the effect of diameter, the rate of loss falls at first rapidly as the diameter increases, but the variation becomes small for cylinders above 4 in. in diameter.

Combined Hoist and Air Compressor

By B. J. Lowe

At a certain mine compressed air is used to operate the brake on a single-drum, direct-motion, Corliss winding engine handling but one cage for elevating men from the mine 1000 ft. below the surface. The engine had originally been steam-driven, having two 18x48-in. high-pressure cylinders. An 8-ft. drum was mounted on the shaft. The engine has been remodeled and is now driven by a 720-hp. geared motor. The gear teeth are of the double-herringbone type and have a ratio of 4.5 to 1.

The pinion is made of nickel steel and the spur wheel of cast steel, the shaft of which couples direct to the drum shaft by a "ham" coupling. One cylinder of the engine is used, the crosshead and connecting-rod being coupled to the crank. On the coupling side of the drum the crank has been removed and a crank disk substituted which contains the male side of the coupling. The motor is operated by three-phase alternating current.

In lowering the men, the post brake is released and the cage descends of its own weight to the bottom of the shaft. While the rope and drum are paying out the steam cylinder is turned into an air compressor, as the valve-gear is still on the cylinder side and the reversing link is set against the engine. The exhaust valves therefore are the admission valves and the former admission valves are the exhaust valves.

The air is compressed against a special relief valve under the operators' platform, which is operated by a lever. The operator controls the speed of the drum by opening and closing this relief valve, reducing or increasing the air pressure against the engine piston. The new arrangement is clearly illustrated in Fig. 1.

The brake path of the drum wore out of true, and a special stand was rigged up and fastened to the brake post. Upon this stand a rock-drill cradle and cylinder were placed to act as a slide rest, and upon the cylinder an electric emery grinder was fastened. To operate, the motor driving the drum is started and then the grinder is set to work truing up the brake path. The arrangement is shown in Fig. 2. This made a very good job and gave a better gripping effect for the brake blocks, which is very essential on a winding engine raising or lowering men.

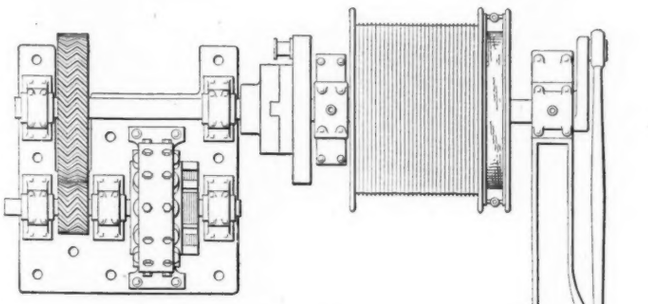


FIG. 1. NEW ARRANGEMENT OF WINDING ENGINE

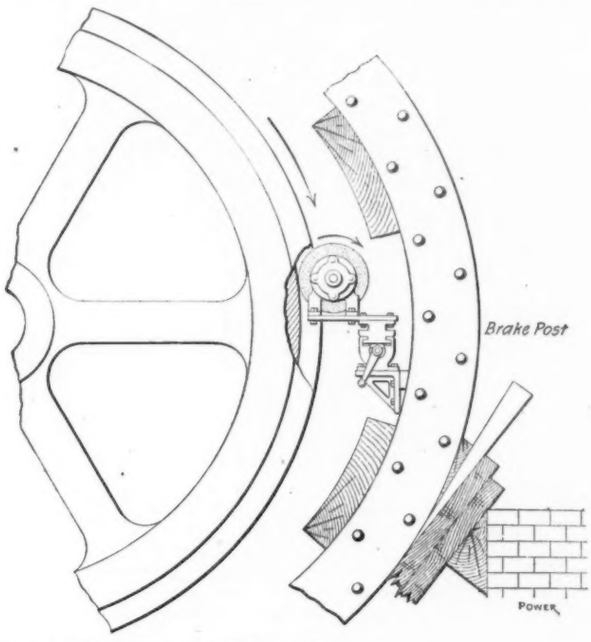


FIG. 2. TURNING THE BRAKE RIM

The Meklong Railway Co., Ltd., of Siam, is about to experiment with an engine burning both firewood and oil.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Use of Back-Pressure Valve

Can exhaust steam be used for supplying heating coils of a drying room without having an exhaust back-pressure relief valve?

B. L.

A back-pressure relief valve should be employed to regulate delivery of steam of sufficient pressure to the heating coils and to relieve the exhaust of unnecessarily high back pressure.

Hard Drawn vs. Annealed Wire

What is the relative resistivity of hard-drawn copper wire and of annealed copper?

J. D.

The resistivity of hard-drawn copper wire is about 2.7 per cent. greater than that of annealed copper.

Evaporation from and at 212 Degrees

What is meant by equivalent evaporation from and at 212 deg.?

G. S.

The relative quantity of water that would be evaporated if the same amount of heat had been absorbed only as latent heat of vaporization at 212 deg. F., and under atmospheric pressure.

Calculating Length of Belt

Can you give a practical formula for calculating the length of an open belt, knowing the distance between centers and diameters of the pulleys?

R. A.

Where

D = Distance between centers;
R = Radius of the larger pulley;
r = Radius of the smaller pulley;
x = Required length of belt;

all dimensions being taken either in feet or in inches, a close approximation to the true length is given by the formula

$$x = 2D + \pi (R + r) + \frac{(R - r)^2}{D}$$

Bright Work on Engines

What are the practical advantages of keeping engine parts bright?

J. C.

Heat radiates slower from highly polished surfaces than from dull or rough surfaces; therefore heads and other uncovered parts of engine cylinders should be kept highly polished to hinder cylinder condensation and obtain higher economy of steam. All working parts of an engine should be kept bright and clean by frequent grooming, not only for good appearances, but for early discovery of any defects which may have developed in the material or adjustment of working parts.

Horsepower Constant

What is the horsepower constant of an engine having piston 20 in. diameter, with stroke of 40 in. and running at 70 r.p.m.?

D. P.

The area of the piston would be $20 \times 20 \times 0.7854 = 314.16$ sq.in. and with the stroke of 40 in. and 70 r.p.m., the piston speed would be

$$\frac{40}{12} \times 2 \times 70 = 466.66 \textit{ ft. per min.}$$

As the horsepower constant is the number of horsepower that would be developed per pound of mean effective pressure, it is the same as the number of horsepower developed by 1 lb. mean effective pressure. The horsepower constant of the given engine, therefore, is

$$\frac{1 \times 314.16 \times 466.66}{33{,}000} = 4.443$$

Stopping a Boiler Leak

What is the best way to stop a leak which appears to extend for 2 to 3 in. along one of the girth seams on the bottom of a 66-in. return-tubular boiler which is in first-class condition in other respects, if calking the leak has proved to be only a temporary remedy?

J. P.

Failure to hold after careful calking may indicate that the leak is due to greasy feed water. Feed about a half pound of soda ash per hour during several days running, then wash the boiler out with clean water, scrape away the former calking and recalk lightly. When the boiler is refilled add 5 to 10 lb. of cornstarch. Further light calking may be necessary, but if a small leak cannot be taken up in the manner described, it may be due to folds of the shell plate which can only be made tight by removing a number of rivets on each side of the leak, drawing the plates close together by bolts and pounding, and replacing the bolts by a good job of snap riveting and calking.

Power to Drive an Ammonia Compressor

What would be the horsepower required for a double-acting ammonia compressor, 15x30 in., 65 r.p.m., with a suction pressure of 18 lb. and a condenser pressure of 200 lb.?

H. R. J.

The refrigerating capacity of the compressor may be expressed by the equation

$$T = \frac{S \times D \times E}{F}$$

in which

T = Refrigerating capacity of compressor per 24 hr.;
S = Piston speed in feet per minute = 325 ft.;
D = Apparent displacement in cubic feet for 1 ft. of piston travel = 1.227;
E = Displacement efficiency of the compressor = 80 per cent.;
F = Cubic feet of gas per minute per ton per 24 hr. = 3.82.

Substituting these values in the formula gives

$$\frac{325 \times 1.227 \times 0.80}{3.82} = 83.5 \textit{ tons}$$

and taking the horsepower per ton as 1.65 would make

$$83.5 \times 1.65 = 137.77 \textit{ hp.}$$

or approximately 140 hp.

Accumulation of Back-Pressure

Where exhaust steam from a pair of engines is used for heating factory buildings, what may be the cause and remedy for occasional failure of the balanced back-pressure valve to relieve excessive back pressure?

J. N.

The trouble is likely to occur where the back-pressure valve is at the foot of an exhaust pipe which has its surface exposed to the weather. Steam leaking through the back-pressure valve after condensing in the exhaust pipe above the valve, falls to the valve and helps to weigh it down. If the exhaust is then rejected by the heating system, back pressure increases, finally causing the engine to slow down unless sufficient back pressure to raise the valve has sooner accumulated. A remedy is the simple one of relieving the condensation formed above the back-pressure valve by a small bleeder, which should be protected from frost.

Study Questions

This Week's Questions
Last Week's Answers

(111) If the resistance of 1000 ft. of copper wire at 90 deg. F. is 100 ohms, what will be the resistance per foot at 75 deg. F.?

(112) If an engineer is now one and a third times as old as he was when he took charge of his present plant, and an oiler, who was born five years before the engineer took charge, is now two years older than half as old as the fireman was when the engineer took charge; how old is each now if the sums of their ages, when the older is as old as the engineer was when he took charge, will be 150 years?

(113) Steam enters a jet condenser at 16 lb. abs. pressure. The temperature of the injection water is 48 deg. F. If the temperature of the discharge water is 115 deg. F., how much injection water (pounds) is necessary per pound of steam?

(114) A hollow shaft has an outside diameter of 3 in., and an inside diameter of 2 in. When twisted by a force of 3000 lb. at a distance of 1 ft. from the center or axis of the shaft, what will be the maximum unit stress developed?

(115) A volume of air under 60 lb. pressure is at a temperature of 350 deg. F. If the temperature is reduced to 32 deg. F., what will be the pressure?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

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(106) The field current will be

$$\frac{E}{r} = \frac{110}{55} = 2\ amp.$$

Therefore the total armature current is 152 amp., and the armature loss is

$$I^2R = (152)^2 \times 0.03 = 693.12\ watts$$

(107) Four volumes may be considered—that of the water in the vessel (let it equal V_1), that of the ball (V_2), that of the frustum of a cone circumscribing the ball (V_3) and that of the frustum of a cone left beneath the ball (V_4).

By the conditions of the problem

$$V_1 + V_2 = V_3 + V_4$$

The formula for the volume of the frustum of a cone of revolution is

$$V = \tfrac{1}{3}\pi H\ (R^2 + r^2 + Rr)$$

where

$\pi = 3.1416$;
H = Height of the frustum;
R = Radius of one base;
r = Radius of the other base.

From this the volume of the vessel is

$$V = \tfrac{1}{3} \times 3.1416 \times 10\left[\left(\frac{9}{2}\right)^2 + \left(\frac{9}{4}\right)^2 + \left(\frac{9}{2} \times \frac{9}{4}\right)\right] = 371.1015\ cu.in.$$

A quarter of its contents (the volume of the water) will be

$$V_1 = \frac{V}{4} = 92.7754\ cu.in.$$

The formula for the volume of a sphere is

$$V = \frac{4}{3}\pi r^3$$

where

$\pi = 3.1416$;
r = Radius of the sphere.

The ball's volume then will be

$$V_2 = \frac{4}{3} \times 3.1416 \times r^3 = 4.1888\, r^3$$

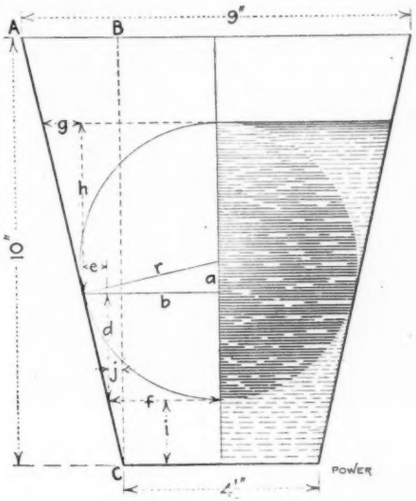
and r is the unknown radius to be determined.

The volume of the circumscribing frustum about this sphere will be

$$V_3 = \tfrac{1}{3} \times 3.1416 \times 2r\ [(b + g)^2 + f^2 + (b + g)\ f]$$

where (referring to the diagram)

$b + g$ = Radius of the top base;
f = Radius of the bottom base.



These three quantities can be expressed in terms of r by the relations of similar triangles as follows:

In the triangle $ABCA$ the side AB will be the difference between the radii at the top and bottom of the vessel, or 2¼ in.; the side BC is the height, 10 in., and the side AC is

$$\sqrt{AB^2 + BC^2} = \sqrt{\left(\frac{9}{4}\right)^2 + 10^2} = \frac{41}{4} = 10\tfrac{1}{4}\ in.$$

Then the radius of the sphere r, as drawn perpendicular to the side AC where the sphere is tangent to the vessel, and the line b perpendicular to BC (half a chord between opposite points of tangency) and a, a perpendicular to this chord passing through the center of the sphere, form a triangle similar to $ABCA$. Hence

$$r : AC :: b : BC$$

or

$$r : \tfrac{41}{4} :: b : 10$$

$$b = \frac{10r}{\frac{41}{4}} = \frac{40r}{41}$$

Also by similar triangles

$$g : a :: h : b$$

or

$$g = \frac{ha}{b}$$

but, by construction

$$h = r + a$$

Again by similar triangles

$$r : AC :: a : AB$$

$$r : \tfrac{41}{4} :: a : \tfrac{9}{4}$$

$$a = \frac{\frac{9}{4}r}{\frac{41}{4}} = \frac{9r}{41}$$

Then

$$h = r + \frac{9r}{41}$$

and

$$g = \frac{\left(r + \frac{9r}{41}\right)\left(\frac{9r}{41}\right)}{\frac{40r}{41}} = \frac{45r}{164}$$

Therefore

$$b + g = \frac{40r}{41} + \frac{45r}{164} = \frac{205r}{164}$$

By construction

$$f = b - e$$

and by similar triangles

$$e : a :: d : b$$

or

$$e = \frac{ad}{b}$$

a and b have already been determined in terms of r and by construction

$$d = r - a = r - \frac{9r}{41}$$

hence

$$e = \frac{\left(\frac{9r}{41}\right)\left(r - \frac{9r}{41}\right)}{\frac{40r}{41}} = \frac{36r}{205}$$

and

$$f = \frac{40r}{41} - \frac{36r}{205} = \frac{164r}{205}$$

Substituting now in the equation for V_3

$$V_3 = \tfrac{1}{3} \times 3.1416 \times 2r\left[\left(\frac{205r}{164}\right)^2 + \left(\frac{164r}{205}\right)^2 + \left(\frac{205r}{164} \times \frac{164r}{205}\right)\right]$$

which reduces to

$$V_3 = 6.7073r^3$$

The volume of the frustum below the sphere is

$$V_4 = \tfrac{1}{3} \times 3.1416 \times i\left[f^2 + \left(\tfrac{9}{4}\right)^2 + \left(f \times \tfrac{9}{4}\right)\right]$$

f is already known in terms of r. By similar triangles

$$i : b :: j : a$$

$$i = \frac{bj}{a}$$

b and a are known in terms of r and, by construction

$$j = f - 2\tfrac{1}{4}$$

hence

$$i = \frac{\left(\frac{40r}{41}\right)\left(\frac{164r}{205} - \frac{9}{4}\right)}{\frac{9r}{41}} = \frac{1312r}{369} - 10$$

Therefore

$$V_4 = \tfrac{1}{3} \times 3.1416\left(\frac{1312\,r}{369} - 10\right)\left[\left(\frac{164\,r}{205}\right)^2 + \left(\frac{9}{4}\right)^2 + \left(\frac{164\,r}{205} \times \frac{9}{4}\right)\right]$$

which reduces to

$$V_4 = 2.383r^3 - 53.0145$$

Now substituting in the original equation

$$V_1 + V_2 = V_3 + V_4$$

$$92.7754 + 4.1888r^3 = 6.7073r^2 + 2.383r^3 - 53.0145$$

This reduces to

$$r^3 = 29.7439$$

and extracting the cube root

$$r = 3.0984$$

or approximately 3.1 in. The diameter of the ball is therefore 6.2 in.

108) The gage pointer indicating 9 in. of vacuum shows that the pressure within the condenser is that amount less than atmospheric pressure. As 1 in. of mercury corresponds to 0.49 lb. per sq.in. and atmospheric pressure is approximately 14.7 lb., the condenser pressure is

$$14.7 - (0.49 \times 9) = 10.29$$

or approximately 10¼ lb.

(109)

$$I.hp. \times Efficiency = B.hp.$$

The total steam consumption = 28.6 × i.hp., or

$$28.6 \times \frac{B.hp.}{E} = \frac{28.6 \times B.hp.}{0.87}$$

Then the steam consumption per brake horsepower would be

$$\frac{28.60}{0.87} = 32.87\ lb.$$

(110) The thermal efficiency is the ratio of the heat utilized to the heat supplied. If all the heat taken from the steam were converted into work the efficiency would be

$$\frac{T_1 - T_2}{T_1}$$

where

T_1 = Initial absolute temperature;

T_2 = Final absolute etemperature.

From steam tables the temperature of steam at 85 lb. abs. is 316.3 deg. F. and at 2 lb. abs. 126.15 deg. F. To convert these to absolute temperatures 461 deg. must be added to each. Then the efficiency will be

$$\frac{(316.3 + 461) - (126.15 + 461)}{(316.3 + 461)} = \frac{190.15}{777.3} = 0.2446$$

or nearly 24½ per cent.

The energy from Niagara Falls, including operation on both sides of the falls, is used at the rate of 126,000 hp. for electro-chemical processes 56,200 hp. for railway service, 36,400 hp. for lighting and 54,500 hp. for various industrial services, the total being 273,140 hp. Since the water of Niagara Falls represents probably 5,000,000 hp., it would seem that only about 5.5 per cent. of the available power is being utilized at present.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

A railroad company saves $81 a year by reducing the length of its pins one-sixteenth inch. When catching the point of this small economy we remember that the last pin which entered our system was about that much too long.

HORAN—Th' paapers sez we've no official standard fer drinkin' wather in these U. S.

DORAN—They're roight, Terence. Ivry toime Oi gets a sthick in moine, Oi puts me foot in ut.

Power-plant proverb: Into the exterior darkness cast thy whinings and grumblings. Then thy days will be long in comfort of toil, thy fellows at peace with thee, and thy boss more satisfied.

Back to the soil! is all very well for the man who wants to do farming, but there's 57 varieties of "soil" in a power plant a clean man will back away back from.

Let's have a "hall of fame" for those men in our field whose names hit off their vocation, or are at variance with it—like W. G. Snow, an authority on heating and ventilation. Mr. Snow caps the peak of this lofty mountain of unknown talent. Enter your neighbor if he is eligible, and POWER readers will soon have a hall of fame that will make the Singer Building look like a bungalow.

"Prosperity is the lot of the coal trade," says a coal paper. The reader sending the most appropriate comment on this item, omitting profanity where possible, can have it published gratis.

A twice-told tale worth repeating: An owner in checking up his bill for the equipment just installed in his newly acquired power plant, complained that 120 pieces were still missing. "I've got all but the 120 volts you said would go with the generator," he wrote the manufacturers.

Hex Nutt says that there's lots in a name when it has a nutty flavor. Now he's afraid that owing to reports of an early winter the squirrels will get him.

At last Europe is afforded a chance to reduce friction and pour oil on the troubled waters. Nearly 142,000,000 gal. of lubricating and paraffin oil have been imported from the United States so far this year.

Smoke Abatement Meeting

That engineers are still interested in smoke abatement, popular opinion to the contrary notwithstanding, is shown by the number of meetings held by engineering bodies lately. Within a month there were three—one the meeting of the International Association for the Prevention of Smoke at Indianapolis. The other two were more local in character. In Philadelphia, Oct. 5, was a joint meeting of the members of the American Society of Mechanical Engineers in that city with the Engineers Club of Philadelphia.

Oct. 8, the New York membership of the American Society of Mechanical Engineers held a very enthusiastic public meeting, attended by some 300, in which various municipal and technical associations and clubs were invited to participate. The topic was introduced by George H. Perkins, head of the engineering department of the Lowell Textile School, who gave a report of the International Smoke Abatement Exhibition and Conference held in London last March.

From this outline of the work that is being done abroad, the attention of the meeting was turned to this country. Several who have been "on the firing line" in every sense of the word, in the battle to reduce the smoke nuisance in some of our Western cities gave interesting accounts of this work and what was accomplished. Among these were P. P. Bird, formerly smoke inspector of Chicago, and a written discussion from O. Monnet, now holding that position, was also read. The problem from the municipal standpoint was further discussed by O. P. Hood, chief mechanical engineer of the Bureau of Mines, Washington, D. C.; Lieut. James Reed, Jr., U. S. N., assistant director of Public Works, Philadelphia, and John M. Lukens, chief of the Bureau of Boiler Inspection, Philadelphia.

Prof. L. P. Breckenridge, Yale University, who, as head of the Engineering Experiment Station at the University of Illinois, did some signal work in the study of the smokeless combustion of soft coal, contributed a very pertinent discussion dealing not only with the problem as related to the city, but from its scientific side as well.

Others that spoke on the engineering phase, pointing out the causes of smoke production and means for preventing it were Albert A. Cary, George L. Prentiss, William Kent, Prof. D. S. Jacobus and A. A. Adler.

Written discussions had been offered by several who could not be present, but the lateness of the hour prevented their being read. These contributions will be digested in the Society's *Journal* later. Among their authors were Charles T. Main, Boston, Mass.; Horace C. Porter, Experiment Station, Bureau of Mines, Pittsburgh; Prof. J. M. Moyer, Pennsylvania State College; Perry Barker, Arthur

D. Little, Inc., Boston; H. V. Coes, Lockwood, Greene & Co., Chicago; Prof. C. H. Benjamin, Purdue University; E. C. Fisher, Wickes Boiler Co., Saginaw, Mich.; C. S. Davis, William B. Pierce Co., Buffalo, N. Y.; George Henderson, Baldwin Locomotive Works, Philadelphia; J. S. Shoemaker, Cumberland Mills, Maine; E. A. Uehling, Uehling Instrument Co., Passaic, N. J., and M. C. Huyette, Murphy Iron Works, Buffalo, N. Y.

Fatality at Worcester, Mass., Power Plant

On Oct. 9, a boiler tube in the power plant of the Worcester Consolidated Street Ry. blew out, killing Arnold S. Allen, the chief engineer, and injuring two firemen.

Nothing more is known of the accident, as the dispatch was received when the forms for this issue were being closed up.

Explosion on Destroyer "Walke"

An explosion on the destroyer "Walke," on Oct. 1, killed three men and injured six. The boat was making speed trials in a heavy sea off Brenton's Reef Lightship, near Newport, R. I., when, it is reported, her port turbine blew off its head casing and steam chest.

Perfect discipline was maintained, relief measures were instituted promptly, and the destroyer was able to reach port unassisted, running at half speed, driven by her starboard turbine, which was uninjured.

OBITUARY

William H. Corbin

On Sept. 25, William Horace Corbin, vice-president and director of the Joseph Dixon Crucible Co., Jersey City, N. J., died at his country home at the Hartwood Club, Sullivan County, N. Y., after an illness of several weeks.

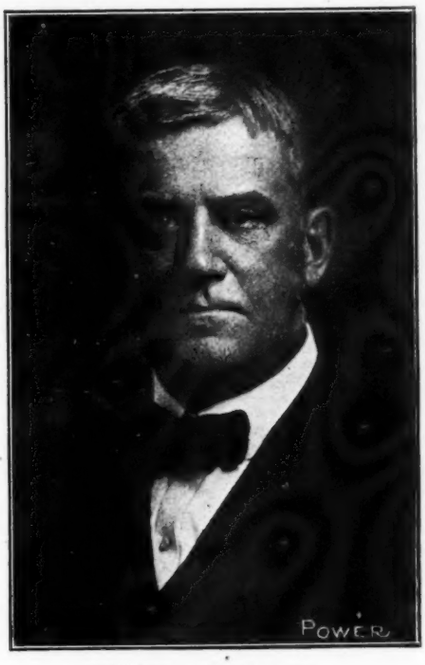
Mr. Corbin was born in McDonough, Chenango County, N. Y., July 12, 1851, the son of Eli L. Corbin and Abigail (Taintor) Corbin. He was educated at Cornell University and Columbia College Law School, from which he was graduated in 1872 and admitted to the New York bar. He came to Jersey City, and was admitted to the New Jersey bar in 1874, making his home in Elizabeth.

For several years, Mr. Corbin was the managing clerk of Senator Brinkerhoff, but in 1881 became a member of Collins & Corbin, which had been formed in 1875 by his brother Charles and Gilbert Collins. That partnership continued until the death of Charles L. Corbin in August, 1911, the firm continuing until W. H. Corbin's death.

Mr. Corbin occupied many positions of honor and responsibility in city and state affairs, helped revise the general corporation act of 1896, and was the author of an annotated edition of that law.

He was active in organizing several large corporations and was a member of the Union League Club of New York, the Union League Club of Hudson County, the Lawyers' Club and the Railroad Club of New York, the Town and Country Club of Elizabeth and the Carteret Club of Jersey City.

Socially, Mr. Corbin was a genial and pleasant companion and deeply religious. Mr. Corbin married Miss Clementine Kellogg, daughter of the late Elijah Kellogg, of Elizabeth. He is survived by his



William H. Corbin

widow and two sons, Clement K. Corbin, of the firm of Collins & Corbin, and Horace K. Corbin, an engineering contractor.

The Jersey City *Journal* said of Mr. Corbin: "In spite of the demands on his time made by good works, he was diligent in business and leaves a competence for his family, besides the heritage of a good name and the sympathy of the entire community."

SOCIETY NOTES

At a meeting of the board of governors of the American Society of Heating and Ventilating Engineers, Sept. 28, the following were appointed to coöperate with a committee of the National District Heating Association on educational matters in heating and ventilation: E. F. Capron, Chicago, Ill.; D. M. Quay, Cleveland, Ohio; W. F. Davendorf, Rochester, N. Y.; M. L. Foote, Columbus, Ohio, and W. C. Green, Cincinnati, Ohio.

The annual meeting of the American Society of Engineer Draftsmen, held in Teachers College, Columbia University, on Tuesday, Oct. 1, was the most important convention yet held by the organization and much was accomplished.

In his opening remarks the president displayed a unique chart showing that the society had quadrupled in the past year. Among other evidences of progress was cited the fact that members had secured, by voluntary subscription, the latest pattern stereopticon projectograph for its monthly meetings and for preparing illustrated lectures to circulate among its branches. A construction program was adopted for the coming year, which included the establishment of a mutual benefit section, involving a form of insurance.

The following officers were elected for 1913: Prof. Charles William Weick, of Columbia University, president; William B. Harsel, first vice-president; Charles A. Clark, second vice-president; C. B. J. McManus, third vice-president; L. T. Maenner, fourth vice-president; E. F. Chandler, Henry L. Sloan and C. W. Fleming, board of governors. The enthusiasm shown by members the past year forms the very best augury for a still more rapid expansion in the year to come.

PERSONAL

J. L. Hopkins will manage the Pittsburgh branch recently opened in the Westinghouse Bldg. by the Durabla Manufacturing Co., New York City.

Louis M. Pawlett has resigned as executive engineer for the Locomobile Co. of America, to devote himself to practice as a specialist in consulting and development work in all departments of the field of motor-transportation engineering. His offices are at 1786 Broadway, New York City.

J. McA. Duncan has been appointed district manager of the Westinghouse Electric & Manufacturing Co. for the Pittsburgh district in place of W. F. Fowler, who has resigned to accept a position with the W. S. Kuhn Corporation. Mr Duncan has been in the employ of the Westinghouse Electric & Manufacturing Co. for about 25 years, and is one of the original group of eight men taken from the Union Switch & Signal Co. to form the electric company.

Soft metals, such as britannia-metal or antimonial-lead, should not be cleaned in the same kettle as that used for removing the grease from brass goods. Tin and lead, contained in the soft metals, are dissolved by the potash or soda in the alkali, and the brass will then become coated. Separate kettles should be used.—*Brass World.*

Moments with the Ad. Editor

Lamoni, Iowa, and Utica, N. Y., are two widely separated places. If we were back in the good old stage coach days, a journey from one of them to the other would be an undertaking too formidable for any ordinary man.

One community might be wiped off the map by an earthquake and it would be months before the other would know about it.

Nowadays, thanks to the railroad, the telegraph and the newspaper, these places are, figuratively speaking, next door neighbors.

Each gets in touch with the other and gets the other's viewpoint.

Two engineers, for instance, through Power, are getting the self same advanced information on power plant practice.

They get the same view of advertising, too; if you doubt that, read their letters—the first is from Roscoe Willey of Lamoni:

Your request of a few weeks ago for letters telling what we bought through the selling section of Power has interested me in that it gives me satisfaction to tell the public what I've found to be good goods.

I take Power as much for the ads as for the help it gives on plant operation. When it arrives I start at the front cover and go straight through before stopping, as I am anxious to know what has happened in the other fellows' experiences and what new device has come out.

When the supply salesman comes around I always know what I want and I tell him if his house does not carry it, I prefer to order direct from the maker rather than to try "something just as good."

Give me a manufacturer that's not afraid to make a noise about his product and offers you a free working sample or to install apparatus on trial for approval and he's the man for me.

I have, through samples, obtained a certainty that could be obtained no other way.

I want to head my list with the "Philadelphia Compressed Air Grease Cup." That's the best cup made.

McClave Shaking Grate, Albany Grease, Palmetto Packing, Jenkins Bros. Globe Valves, American Gauges, Moncreiff Gauge Glasses, Dean Boiler Tube Cleaner, two Murray Corliss Engines, Dixon's Flake Graphite, Kewanee Unions, I. C. S. Reference Books, McGraw-Hill books and Smooth-on Cement were called to our attention through Power.

We are to be in the market for mechanical stokers and a new boiler before long, and you bet we are awake studying that class of goods.

And Wallace Johnson of Utica differs from his Iowa brother only in the details of what his concern has bought through Power:

From my point of view the advertising pages in Power are of as much value to the engineer as any part for it is there we find what new and up to date things are to be had.

I think most all of us can remember when we have gone to a lot of trouble and expense to make something to get along with and then found out that the same thing much better could be bought had we known about it.

I am now using a Bury Air Compressor that I saw described in Power and it was just the very thing needed to fill the place, and I looked back a long way to find that particular issue of the paper.

I have also an American-Thompson Indicator from the American Steam Gauge & Valve Mfg. Co., and two reducing wheels; two Manzell Cylinder pumps; one Lagonda Tube Cleaner; Garlock packing everywhere. And last, but not the least by any means, a Westinghouse Leblanc Condenser, now taking care of about 1600 to 1800 h.p. I first saw this condenser and air pump advertised in Power and it looked good to me. All these I read about and now have and every one great and small is giving splendid satisfaction.

* * *

Yes, it pays to read Power advertising and buy goods advertised in Power.